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RESEARCH MEMORANDUM

for the

Bureau of Ordnance, Navy Department

INVESTIGATION OF TURBINE OF MARK 25 TORPEDO POWER PLANT

WITH FIVE NOZZLE DESIGNS

By Jack W. Hoyt and Harry Kottas

Flight Propulsion Research Laboratory
Cleveland, Ohio

CLASSIFIED DOCUMENT

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INVESTIGATION OF TURBINE OF MARK 25 TORPEDO POWER PLANT

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SUMMARY

Efficiency investigations were made on the two-stage turbine from a Mark 25 aerial torpedo to determine the performance of the unit with five different turbine nozzles. The output of the turbine blades was computed by analyzing the windage and mechanical-friction losses of the unit. A method was developed for measuring the change in turbine clearances with changed operating conditions. The turbine was found to be most efficient with a cast nozzle having a sharp-edged inlet to the nine nozzle ports.

INTRODUCTION

Torpedoes operating on a combustion cycle require a high-pressure-ratio gas turbine to drive the propellers of the unit. Some types of rocket may employ similar turbines to operate fuel pumps. Both applications involve the extraction of maximum power over a short time with minimum size and weight of the power plant and the fuel load. Because of the need for information on turbines of this type, at the request of the Bureau of Ordnance, Navy Department, an investigation is being conducted at the NACA Cleveland laboratory of a two-stage turbine from an aerial torpedo to determine the performance of the unit under steady-state conditions with various types of turbine nozzle and to compare the effect of the nozzle designs on the over-all turbine efficiency.

A series of five turbine nozzles was investigated to determine the effect of pressure ratio, blade-jet speed ratio, and nozzle design on turbine efficiency. This investigation covered a range of pressure ratios from 8 to 20 and turbine speeds from

6000 to 18,000 rpm with an inlet-gas temperature of 1000° F and an inlet-gas pressure of 95 pounds per square inch gage. The true output of the nozzle-and-turbine-blade combinations was determined by evaluating the power losses of the turbine due to windage and mechanical friction. A method was developed for measuring the change in turbine clearances with changed turbine operating conditions, and the effect of nozzle-wheel clearance on turbine efficiency was investigated.

APPARATUS AND INSTRUMENTATION

Turbine. - The Mark 25 torpedo power plant is a two-stage counterrotating impulse turbine with integral speed-reduction and power-equalizing gearing. The power-output shafts are also counterrotating and operate at 0.099 turbine speed. A combining gearbox was specially designed for this installation to convert the dual rotation of the torpedo power-output shafts into single-direction rotation.

A sketch of the nozzle-and-turbine-wheel assembly is shown in figure 1. The gas expands through the partial-admission nozzles and enters the forward turbine blades at supersonic relative velocity. Because considerable gas-velocity energy is absorbed by the forward turbine, the flow leaving the first rotor and entering the counterrotating rear wheel is subsonic. Although a small amount of reaction is employed in the rear turbine to extract maximum energy, the rear turbine is included in this design mainly to eliminate gyroscopic effects. The shroud-band diameters of the forward and rear turbine wheels are 11.000 and 11.320 inches, respectively.

Nozzles. - The high inlet-gas temperatures and pressures, the high pressure ratios, and the low gas weight flows for this turbine necessitate special care in the design and the construction of the turbine nozzles. The high pressure ratios in the nozzles cause supersonic flow, and very small nozzle throats and partial admission are necessary for low weight flow.

The nine-port nozzles (A, E, G, and H) have 90° nozzle-arc gas admission, whereas the three-port nozzle F has 30° nozzle-arc gas admission. Nozzles A and E are part of a series of nozzles investigated under actual torpedo conditions at Massachusetts Institute of Technology (reference 1). Nozzles F, G, and H are a continuation of this nozzle series.

Nozzle A has rounded inlets to the nine rectangular, converging-diverging nozzle ports; the nine nozzle throats are approximately 0.25 by 0.10 inch. The nozzle was produced by the precision lost-wax casting method and has hand-filed nozzle ports.

The difficulty in manufacturing nozzles similar to nozzle A led to the design of a nozzle that could be more easily fabricated. Nozzle E (fig. 2(a)) has reamed ports with rounded inlets. The ports are cylindrical with no area divergence; the necessary expansion takes place in the axial-clearance space between the nozzle and the forward turbine buckets. The nine nozzle ports are approximately 0.163 inch in diameter. Nozzle F is similar to nozzle E except that the six ports nearest the gas inlet are blocked by a welded plate, leaving only three ports in use. The nozzle ports are also 0.163 inch in diameter.

In order to provide a more favorable flow profile and to prevent the combustion gas from spilling above or below the forward-wheel blades, the shrouded nine-port nozzle G (fig. 2(b)) was designed with the same nozzle dimensions as nozzle E but with a shroud projecting axially 0.216 inch from the outlet face.

Nozzle H (fig. 2(c)) was produced by refinement of both design and casting technique to give more accurate dimensional control and a smoother nozzle surface. This nozzle has nine rectangular ports with sharp inlet edges and throat sizes of approximately 0.25 by 0.10 inch.

Setup and instrumentation. - The setup used for this investigation is shown in figure 3. A 300-horsepower electric dynamometer was used to absorb the turbine output and to drive the turbine during motoring runs. The dynamometer speed was measured with a chronometric tachometer. Dynamometer torque measurements were made with a scale having a range of 0 to 250 pounds.

Combustion gases at a pressure of 95 pounds per square inch gage were furnished by a burner using unleaded gasoline and air. The altitude exhaust at the turbine outlet was used to obtain the desired range of pressure ratios across the turbine. The air flow was metered by a standard A.S.M.E. orifice in the inlet-air pipe. The fuel flow was measured by means of a calibrated rotameter.

Gas-stream temperatures were measured by triple-shielded thermocouples of 0.50-inch outside diameter and 0.75 inch long. The inlet-gas temperatures were taken by two thermocouples mounted in the insulated inlet-gas pipe. Two thermocouples were located in the exhaust cone 2 inches behind the rear turbine wheel. Seven

static-pressure taps were placed on the outer heat-shield ring to provide a pressure survey along the nozzle arc above the wheels in order to evaluate the gas density around the turbine. Static and total pressures were taken in the inlet- and outlet-gas pipes. In addition, a total-pressure tube insensitive to yaw was mounted in the exhaust cone 2 inches behind the rear turbine wheel in the same plane as the exhaust thermocouple.

The turbine was modified for long-duration efficiency runs by removing the integral oil pump on the unit and installing an external oil system with instruments to measure the oil temperature and the flow rate. Thermocouples were installed on the four high-speed bearings to guard against failures by overheating of the bearings. Thermocouples were also placed on the front and rear oil seals nearest the turbine wheels to indicate any hot-gas leakage past the seals into the turbine bearings. Sufficient temperature measurements were taken on the aluminum housing and cover of the unit to avoid operation above the safe-temperature limits of these parts. Thermocouples were located on the stainless-steel heat shields to indicate the operating temperatures.

During runs to determine the change in axial nozzle-wheel clearance and radial nozzle setting with operating conditions, clearance indicators were installed on the turbine as shown in figure 4. The method of operation of the axial-clearance indicators is illustrated by figure 5. The gage at the left is shown resting on a lug welded to the turbine nozzle. When the tip of the gage is rotated 180° by turning the handle to the position shown for the gage at the right, the tip is allowed to move past the lug and touch the turbine wheel. The difference between the dial-indicator readings with the indicator tip in the two positions is a measure of the clearance.

Precision. - The precision of the measurements was within the following limits:

| | |
|---------------------------------------|-------|
| Air flow, percent | ±1.00 |
| Torque, foot-pounds | ±0.15 |
| Speed, rpm | ±5 |
| Inlet-gas pressure, percent | ±0.50 |
| Pressure, inches mercury | ±0.05 |

PROCEDURE

The investigation of the Mark 25 torpedo power plant was divided into the following phases:

(1) Power runs to determine turbine efficiency and over-all performance

(2) Motoring runs to evaluate the windage and mechanical losses of the turbine unit for use in efficiency calculations

(3) Clearance studies to find the true running clearance between nozzles and turbine wheel

Power runs. - The over-all efficiency of the turbine power plant and combining gear was determined from the power output with the various nozzles at several pressure ratios and turbine speeds over the range of operation. With the inlet-gas conditions maintained at 95 pounds per square inch gage and 1000° F, the turbine speed was varied from 6000 to 18,000 rpm at pressure ratios of 8, 10, 15, and 20. (Because of air-flow limitations, the maximum pressure ratio obtained with nozzle H was 19.) The values of power output obtained during these runs represent the output of the turbine wheels with the windage and the gear and bearing friction in the power plant subtracted.

Motoring runs. - The rotation losses of the turbine unit due to gears, bearings, and turbine disks were obtained by motoring the unit with disks instead of the turbine wheels. The forward and rear disks were 10.075 and 9.955 inches in diameter, respectively, or exactly the same diameter as the root diameters of the turbine blades. The power required to motor these disks at speeds from 6000 to 18,000 rpm and at different air densities in the turbine case represents the total power absorbed through mechanical losses in the gears and the bearings and through windage of the turbine disks.

The motoring runs were then repeated with standard turbine wheels with forward- and rear-wheel shroud-band diameters of 11.000 and 11.320 inches, respectively. The power then required represents all the losses of the disk unit plus the air-pumping effect of the turbine blades. All motoring runs were made without air flow through the turbine, the turbine nozzle being blocked. Air densities in the turbine case were varied by varying the turbine-outlet-gas pressure.

Clearance studies. - Runs were made at an inlet-gas pressure of 95 pounds per square inch gage, temperatures from 500° to 1500° F, and pressure ratios of 10, 15, and 20 at speeds of 6000 and 12,000 rpm with indicators installed to measure the changes in radial nozzle setting and axial nozzle-turbine clearance with operating conditions. The thermal changes in radial setting and axial clearance were determined by operating the turbine at low speeds with varying inlet-gas temperature. In order to indicate

the change in radial wheel dimensions with centrifugal force, the turbine was motored at speeds of 6000 to 18,000 rpm with sufficient cold air allowed to enter the nozzle inlet to keep the wheels, nozzle box, and turbine case at approximately room temperature so that thermal effects would be eliminated.

Radial setting of the nozzle ports for turbine-efficiency runs with nozzles A, E, F, and G were made to conform to the recommendations of reference 1. Accordingly, a radial setting of 5.283 inches from the turbine-shaft center line to the midpoint of the nozzle port was used for each of these nozzles. Nozzle H, because of its different port design and larger flow area, required experimentation to obtain the best radial setting for the final turbine-efficiency runs. The proper value of 5.258 inches was obtained by over-all turbine-efficiency runs over a range of radial settings from 5.233 to 5.308 inches from turbine-shaft center line to average midpoint of nozzle ports.

The effect of axial clearance on turbine efficiency was investigated by varying the axial nozzle-wheel clearance for nozzle A from 0.030 to 0.090 inch and determining the over-all turbine efficiency for each axial-clearance setting.

Axial nozzle-wheel clearances for turbine-efficiency runs were set, on the basis of the clearance-indicator studies, to give 0.030-inch clearance when the unit reached operating temperature. Because the turbine blades are set back from the edge of the turbine wheel 0.025 inch, the axial clearance from nozzle to blade was thus 0.055 inch.

CALCULATIONS

The isentropic enthalpy drop available for an expansion from the turbine-inlet-gas total temperature and pressure to the outlet-gas static pressure was computed from the air tables of reference 2 and corrected for the effect of the fuel input.

The blade-jet speed ratio is the ratio of the blade velocity at the pitch diameter of the first wheel to the ideal turbine-nozzle jet velocity corresponding to an isentropic expansion from the inlet-gas total temperature and pressure to the turbine-outlet-gas static pressure.

Pressure ratio is the ratio of the inlet-gas total pressure to the outlet-gas static pressure.

Brake efficiency is the ratio of the brake power calculated from the torque and speed of the dynamometer shaft to the isentropic power available; that is

$$\text{brake efficiency} = \frac{\text{brake power}}{\text{isentropic power available}}$$

The mechanical losses of the unit were determined by extrapolating the windage-loss curves to zero air density. The horsepower at zero air density represents the power required to drive the gears and overcome bearing friction and is the mechanical loss for the turbine and the combining gear. The mechanical-loss power added to the brake power gives the power output of the turbine wheels at the shaft; and

$$\text{wheel efficiency} = \frac{\text{brake power} + \text{mechanical losses}}{\text{isentropic power available}}$$

The windage losses were obtained from the windage chart for the nozzle arc of admission under consideration.

The power required to motor the unit with disks installed instead of turbine wheels represents the total power absorbed through mechanical losses in the gears and the bearings plus the windage of the disks. These losses are presented in figure 6(a) and table I. Curves for constant disk speed have been extrapolated to zero air density to separate the mechanical losses from the power losses due to the disk windage. The windage and mechanical losses obtained by motoring the standard turbine wheels are presented in figure 6(b) and table II. Slight differences in mechanical losses in the units shown in figures 6(a) and 6(b) can be ascribed to inherent differences in assembly and to curve extrapolation.

The windage loss for the full blade periphery was calculated by subtracting the disk-windage loss determined from figure 6(a) from the disk-and-blade-windage loss obtained from figure 6(b). Blade-windage loss derived in this manner is considered the loss of the turbine blades with 0° nozzle-arc gas admission.

When the nine-port turbine nozzles with 90° nozzle-arc gas admission (nozzles A, E, G, and H) are used, 90° of the turbine-blade periphery, or one-fourth of the turbine buckets, is active and hence is not subject to windage loss. The windage losses for the unit with these nozzles are composed of the total disk

windage plus three-fourths of the windage due to the turbine blades. Accordingly, figure 7(a) was prepared from the total disk windage plus three-fourths of the additional windage loss due to the turbine blades. This chart can be used to find directly the windage and mechanical losses of the unit with the nine-port nozzles if the gas density in the turbine case is known.

Nozzle F has three ports with 30° nozzle-arc gas admission; hence 30° of the turbine-bucket periphery, or one-twelfth of the turbine blades, is active and thus has no windage losses. Figure 7(b) was prepared by using the total disk windage plus eleven-twelfths of the additional windage loss due to the turbine blades.

The windage-loss power added to the mechanical-loss power and the brake power gives the work output at the turbine blades, and

$$\text{blade efficiency} = \frac{\text{brake power} + \text{mechanical losses} + \text{windage}}{\text{isentropic power available}}$$

RESULTS

Turbine efficiency. - The individual performance of the turbine at an inlet-gas pressure of 95 pounds per square inch gage and temperature of 1000° F and pressure ratios of 8 to 20 with nozzles A, E, F, G, and H are shown in figures 8 to 12 and tables III to VII, respectively. All data shown in figures 8 to 12 were taken with 0.030-inch axial nozzle-wheel running clearance. The maximum brake efficiency of 0.53 was obtained with nozzle H at a blade-jet speed ratio of approximately 0.21 for a pressure ratio of 8 (fig. 12(a)). At the maximum brake-efficiency point, the wheel efficiency (which credits the turbine with the work necessary to drive the gears and bearings) was approximately 0.56. If the turbine is also credited with the work necessary to overcome windage losses, the efficiency was 0.58 at the foregoing conditions. Nozzle H also showed the highest blade efficiency (0.64) of the nine-port nozzles at a blade-jet speed ratio of 0.295 and pressure ratio of 8 (fig. 12(a)) although the peak blade efficiency (which would be at a higher blade-jet speed ratio) could not be determined because of the 18,000-rpm speed limit of the turbine. The higher efficiencies of the turbine with nozzle H is evidence of either (a) lower losses in the nozzle, or (b) more favorable matching of turbine flow area with the greater flow area of this nozzle. Nozzle H has, for example, about 20 percent greater gas mass flow than nozzle A.

The difference in the general trends of the curves of brake efficiency and blade efficiency can be ascribed to the

approximately cubic increase of windage losses with speed. The turbine is therefore heavily penalized by the use of partial-admission nozzles as shown by the three-port nozzle F (fig. 10), for which brake efficiency drops off sharply with increased blade-jet speed ratio but blade efficiency increases with increased speed.

The reamed nozzles generally showed lower efficiencies than the cast nozzles. The addition of a shroud to provide a guide for the nozzle jets resulted in a marked decrease in efficiency (fig. 11), presumably because of increased shock and eddy losses.

The effect of pressure ratio on brake efficiency with nozzle H at an inlet-gas temperature of 1000° F is shown in figure 13(a). The brake efficiency varies only slightly with pressure ratio with this nozzle; over the range of pressure ratios from 8 to 19, the maximum difference in efficiency was 0.01. The effect of inlet-gas temperature on brake efficiency with nozzle A is shown in figure 13(b). An increase in inlet-gas temperature from 500° to 1000° F resulted in an efficiency decrease of 0.03 owing to the increased heat loss from the unit.

Clearances. - The results of clearance-indicator studies of the changes in radial setting and axial clearance between the turbine nozzle and the turbine wheel with changed operating conditions are shown in figure 14 and table VIII. Although the temperatures in the turbine case are affected by turbine speed and pressure ratio as well as the inlet-gas temperature, the changes in setting and clearance were plotted against inlet-gas temperature (which does not reflect those variables) because an average clearance-change value was desired for the turbine operating over a wide range of speed and pressure-ratio conditions for a given inlet-gas temperature. The data plotted in this manner yield a range through which the clearance may vary with a given inlet-gas temperature.

From this chart, a correction of the clearance setting of the turbine may be made before operation to obtain the desired value of running clearance when the parts reach operating temperature. The maximum change in radial setting over the range of inlet-gas temperatures was a decrease of 0.013 inch. The difference between the left and right radial indicators is evidence of a slight tilting of the nozzle. Axial changes in clearance were slightly more pronounced, the maximum decrease being about 0.017 inch. A radial-setting decrease of about 0.005 inch and an axial-clearance decrease of about 0.015 inch were chosen as safe allowances for changes in clearance over a range of inlet-gas temperatures from room temperature to 1500° F.

The results of motoring runs to determine the expansion of the turbine wheels due to centrifugal force are given in table IX. Although the runs were made with room temperature in the nozzle casing, rough calculations indicate that turbine operating temperatures would have only a small effect on radial expansion due to centrifugal force for this turbine. At a turbine speed of about 18,000 rpm with room temperature, the radial expansion was found to be 0.0045 inch.

The variation of brake efficiency with radial setting of nozzle H is shown in figure 15(a). As the radial setting of the nozzle was varied from 5.308 to 5.233 inches, the turbine efficiency increased slightly to a maximum value at a radial setting of 5.258 inches then, with a further decrease in setting, dropped sharply.

The variation of brake efficiency with axial nozzle-wheel clearance of nozzles A and E is shown in figure 15(b). The brake efficiency of nozzle A improved as the running clearance was decreased from 0.090 to 0.030 inch; at 0.030-inch axial nozzle-wheel clearance, the turbine is less efficient with nozzle E than with nozzle A. At a running clearance of 0.090 inch, however, the turbine efficiency with nozzle E is slightly improved over that with nozzle A, which indicates that adequate clearance space must be provided for free gas-jet expansion with a nondivergent nozzle for greatest turbine efficiency.

As a result of the clearance-indicator studies, the axial nozzle-wheel clearance and the radial setting were positioned at 0.015 and 0.005 inch, respectively, more than the running clearances desired during turbine operation.

SUMMARY OF RESULTS

The mechanical and windage losses of a Mark 25 aerial-torpedo power plant were determined and the efficiency of the turbine, based on these loss determinations, was investigated for five turbine nozzles. Clearances were also studied. The following results were obtained:

1. The turbine was found to be most efficient with a cast, sharp-edged-inlet, nine-port nozzle with 90° nozzle-arc gas admission.
2. The maximum brake efficiency of the turbine with the cast, sharp-edged-inlet nozzle at an inlet pressure of 95 pounds per

square inch gage, an inlet-gas temperature of 1000° F, a pressure ratio of 8, and a blade-jet speed ratio of 0.21, was 0.53. If the turbine is credited with the work necessary to drive the gears and the bearings, the efficiency at the foregoing conditions was 0.56. If the turbine is also credited with the work necessary to overcome windage losses, the efficiency was 0.58.

3. Windage and mechanical-loss charts that have been prepared gave the operating loss in horsepower for various turbine-casing pressures, speeds, and nozzle arcs of 30° and 90° .

4. The variation of radial nozzle setting and axial nozzle-wheel clearance with turbine-inlet-gas temperature was small. A radial-setting decrease of about 0.005 inch and an axial-clearance decrease of about 0.015 inch were chosen as safe allowances for changes in clearance over a range of inlet-gas temperatures from room temperature to 1500° F.

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2. Keenan, Joseph H., and Kaye, Joseph: Thermodynamic Properties of Air. John Wiley & Sons, Inc., 1945.

TABLE I - SUMMARY OF DATA FOR WINDAGE AND MECHANICAL LOSSES
FOR MARK 25 TURBINE DISKS

| Turbine speed (rpm) | Horsepower to drive turbine | Air tem- perature in tur- bine case (°F) | Pressure in turbine case (in. Hg abs.) |
|------------------------|-----------------------------------|--|--|
| 6,069 | 0.80 | 134 | 29.36 |
| | .80 | 124 | 24.36 |
| | .76 | 129 | 19.36 |
| | .74 | 132 | 14.36 |
| | .70 | 135 | 9.36 |
| 8,092 | 1.33 | 158 | 29.36 |
| | 1.28 | 159 | 24.36 |
| | 1.20 | 165 | 19.36 |
| | 1.15 | 164 | 14.36 |
| | 1.09 | 163 | 9.36 |
| 10,115 | 2.13 | 187 | 29.36 |
| | 2.07 | 183 | 24.39 |
| | 1.97 | 196 | 19.38 |
| | 1.87 | 194 | 14.40 |
| | 1.77 | 187 | 9.36 |
| 12,138 | 2.96 | 239 | 29.36 |
| | 2.68 | 245 | 24.40 |
| | 2.60 | 247 | 19.36 |
| | 2.48 | 241 | 14.37 |
| | 2.36 | 214 | 9.41 |
| 14,161 | 4.01 | 265 | 29.36 |
| | 3.64 | 275 | 24.42 |
| | 3.41 | 286 | 19.36 |
| | 3.27 | 270 | 14.44 |
| | 3.08 | 229 | 9.99 |
| 16,184 | 5.17 | 304 | 29.36 |
| | 4.69 | 341 | 24.44 |
| | 4.37 | 343 | 19.43 |
| | 4.05 | 330 | 14.46 |
| | 3.89 | 281 | 10.81 |
| 18,207 | 6.48 | 379 | 29.36 |
| | 5.82 | 392 | 24.38 |
| | 5.46 | 389 | 19.40 |
| | 5.04 | 356 | 14.42 |
| | 4.68 | 315 | 10.42 |

TABLE II - SUMMARY OF DATA FOR WINDAGE AND MECHANICAL LOSSES
FOR MARK 25 TURBINE DISKS AND BLADES

| Turbine speed (rpm) | Horsepower to drive turbine | Air temperature in turbine case (°F) | Pressure in turbine case (in. Hg abs.) |
|---------------------|-----------------------------|--------------------------------------|--|
| 6,069 | 1.34 | 182 | 29.30 |
| | 1.26 | 243 | 25.31 |
| | 1.20 | 206 | 18.73 |
| | 1.08 | 163 | 14.32 |
| | .94 | 147 | 9.66 |
| 8,092 | 2.32 | 230 | 29.32 |
| | 2.21 | 237 | 25.45 |
| | 1.97 | 224 | 19.35 |
| | 1.83 | 197 | 14.71 |
| | 1.63 | 179 | 9.58 |
| 10,115 | 3.66 | 306 | 29.36 |
| | 3.33 | 297 | 24.11 |
| | 2.93 | 304 | 19.19 |
| | 2.70 | 299 | 15.26 |
| | 2.33 | 281 | 9.76 |
| 12,138 | 5.53 | 369 | 29.42 |
| | 5.00 | 351 | 23.79 |
| | 4.44 | 369 | 20.15 |
| | 3.84 | 364 | 15.56 |
| | 3.20 | 339 | 9.69 |
| 14,161 | 7.67 | 461 | 29.46 |
| | 6.95 | 461 | 24.90 |
| | 5.83 | 462 | 18.58 |
| | 4.90 | 443 | 13.70 |
| | 4.11 | 416 | 9.41 |
| 16,184 | 10.46 | 558 | 29.54 |
| | 9.65 | 463 | 23.60 |
| | 8.21 | 498 | 18.86 |
| | 6.99 | 481 | 14.02 |
| | 6.03 | 465 | 10.58 |
| 18,207 | 15.83 | 647 | 29.63 |
| | 11.70 | 459 | 15.28 |
| | 9.72 | 432 | 11.20 |
| | 8.10 | 383 | 7.99 |

TABLE III - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE
WITH NOZZLE A

[Inlet-gas temperature, 1000° F; inlet-gas pressure,
95 lb/sq in. gage]

| Pressure ratio | Air mass flow (lb/hr) | Fuel-air ratio | Horsepower available from isentropic expansion | Turbine speed (rpm) | Brake horsepower | Blade-jet speed ratio | Gas density in turbine case (lb/cu ft) |
|----------------|-----------------------|----------------|--|---------------------|------------------|-----------------------|--|
| 8 | 955.2 | 0.0135 | 61.09 | 6,089 | 20.97 | 0.0987 | 0.0354 |
| | 955.9 | .0135 | 61.14 | 8,072 | 24.63 | .1308 | .0358 |
| | 955.9 | .0135 | 61.14 | 10,125 | 27.16 | .1641 | .0362 |
| | 955.9 | .0135 | 61.14 | 12,158 | 28.29 | .1970 | .0369 |
| | 956.6 | .0135 | 61.18 | 14,161 | 28.23 | .2295 | .0370 |
| | 955.9 | .0135 | 61.14 | 16,164 | 27.11 | .2620 | .0369 |
| | 956.6 | .0135 | 61.18 | 18,207 | 23.94 | .2951 | .0367 |
| 10 | 960.3 | 0.0135 | 66.20 | 6,069 | 23.56 | 0.0948 | 0.0294 |
| | 960.3 | .0135 | 66.20 | 8,122 | 27.97 | .1268 | .0301 |
| | 958.2 | .0135 | 66.06 | 10,115 | 30.67 | .1580 | .0302 |
| | 959.3 | .0135 | 66.12 | 12,138 | 32.12 | .1895 | .0304 |
| | 958.2 | .0135 | 66.06 | 14,161 | 32.11 | .2211 | .0304 |
| | 958.2 | .0135 | 66.06 | 16,194 | 31.06 | .2529 | .0305 |
| | 959.2 | .0135 | 66.06 | 18,227 | 28.41 | .2846 | .0303 |
| 15 | 958.2 | 0.0135 | 73.98 | 6,069 | 25.06 | 0.0896 | 0.0210 |
| | 958.2 | .0135 | 73.98 | 8,092 | 30.00 | .1195 | .0210 |
| | 958.2 | .0135 | 73.98 | 10,125 | 33.57 | .1495 | .0213 |
| | 960.1 | .0135 | 74.13 | 12,138 | 35.80 | .1792 | .0209 |
| | 958.2 | .0135 | 73.98 | 14,161 | 36.40 | .2091 | .0211 |
| | 958.2 | .0135 | 73.98 | 16,194 | 35.70 | .2391 | .0215 |
| | 958.2 | .0135 | 73.98 | 18,187 | 33.98 | .2685 | .0220 |
| 20 | 958.2 | 0.0135 | 79.10 | 6,059 | 25.42 | 0.0870 | 0.0167 |
| | 959.3 | .0135 | 79.18 | 8,092 | 30.85 | .1162 | .0169 |
| | 959.3 | .0135 | 79.18 | 10,125 | 34.87 | .1454 | .0167 |
| | 958.2 | .0135 | 79.10 | 12,148 | 37.43 | .1745 | .0166 |
| | 958.2 | .0135 | 79.10 | 14,151 | 38.47 | .2032 | .0165 |
| | 958.2 | .0135 | 79.10 | 16,214 | 37.94 | .2328 | .0169 |
| | 958.2 | .0135 | 79.10 | 18,187 | 36.14 | .2612 | .0175 |

TABLE IV - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE E

[Inlet-gas temperature, 1000° F; inlet-gas pressure,
95 lb/sq in. gage]

| Pressure ratio | Air mass flow (lb/hr) | Fuel-air ratio | Horsepower available from isentropic expansion | Turbine speed (rpm) | Brake horsepower | Blade-jet speed ratio | Gas density in turbine case (lb/cu ft) |
|----------------|-----------------------|----------------|--|---------------------|------------------|-----------------------|--|
| 8 | 976.5 | 0.0138 | 62.49 | 6,099 | 20.96 | 0.0988 | 0.0338 |
| | 977.6 | .0138 | 62.56 | 8,112 | 24.01 | .1315 | .0345 |
| | 977.6 | .0138 | 62.56 | 10,125 | 25.53 | .1641 | .0348 |
| | 976.5 | .0138 | 62.49 | 12,148 | 25.66 | .1969 | .0358 |
| | 976.5 | .0138 | 62.49 | 14,141 | 27.77 | .2292 | .0360 |
| | 977.6 | .0138 | 62.56 | 16,164 | 26.26 | .2620 | .0360 |
| | 976.5 | .0138 | 62.49 | 18,207 | 23.52 | .2951 | .0357 |
| 10 | 977.6 | 0.0138 | 67.42 | 6,079 | 23.00 | 0.0949 | 0.0280 |
| | 978.6 | .0138 | 67.48 | 8,112 | 27.24 | .1267 | .0282 |
| | 977.6 | .0138 | 67.42 | 10,105 | 29.94 | .1578 | .0282 |
| | 977.6 | .0138 | 67.42 | 12,138 | 31.04 | .1895 | .0287 |
| | 977.6 | .0138 | 67.42 | 14,151 | 30.87 | .2210 | .0291 |
| | 977.6 | .0138 | 67.42 | 16,204 | 29.69 | .2530 | .0291 |
| | 977.6 | .0138 | 67.42 | 18,197 | 27.52 | .2842 | .0290 |
| 15 | 977.4 | 0.0138 | 75.50 | 6,079 | 24.22 | 0.0897 | 0.0199 |
| | 976.3 | .0138 | 75.41 | 8,132 | 29.08 | .1200 | .0202 |
| | 977.4 | .0138 | 75.50 | 10,125 | 32.20 | .1495 | .0202 |
| | 976.3 | .0138 | 75.41 | 12,168 | 33.60 | .1796 | .0201 |
| | 974.6 | .0138 | 75.27 | 14,181 | 34.02 | .2093 | .0201 |
| | 976.3 | .0138 | 75.41 | 16,194 | 35.01 | .2391 | .0205 |
| | 974.7 | .0138 | 75.28 | 18,227 | 33.16 | .2691 | .0207 |
| 20 | 975.6 | 0.0138 | 80.56 | 6,059 | 25.58 | 0.0865 | 0.0162 |
| | 975.6 | .0138 | 80.56 | 8,102 | 31.16 | .1157 | .0163 |
| | 975.6 | .0138 | 80.56 | 10,115 | 34.97 | .1445 | .0165 |
| | 976.5 | .0138 | 80.63 | 12,128 | 37.49 | .1732 | .0161 |
| | 976.5 | .0138 | 80.63 | 14,171 | 38.43 | .2024 | .0163 |
| | 976.5 | .0138 | 80.63 | 16,204 | 37.91 | .2314 | .0163 |
| | 976.5 | .0138 | 80.63 | 18,217 | 36.62 | .2601 | .0168 |



TABLE V - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE F

[Inlet-gas temperature, 1000° F; inlet-gas pressure,
95 lb/sq in. gage]

| Pressure ratio | Air mass flow (lb/hr) | Fuel-air ratio | Horsepower available from isentropic expansion | Turbine speed (rpm) | Brake horsepower | Blade-jet speed ratio | Gas density in turbine case (lb/cu ft) |
|----------------|-----------------------|----------------|--|---------------------|------------------|-----------------------|--|
| 8 | 247.3 | 0.0208 | 15.99 | 6,059 | 6.29 | 0.0982 | 0.0354 |
| | 247.3 | .0208 | 15.99 | 8,112 | 6.74 | .1315 | .0358 |
| | 247.3 | .0208 | 15.99 | 10,125 | 6.54 | .1641 | .0355 |
| | 247.3 | .0212 | 16.00 | 12,148 | 5.24 | .1969 | .0356 |
| | 247.3 | .0212 | 16.00 | 14,141 | 3.63 | .2292 | .0348 |
| | 247.1 | .0212 | 15.98 | 16,184 | 1.76 | .2623 | .0338 |
| | 247.3 | .0212 | 16.00 | 18,217 | -.48 | .2952 | .0324 |
| 10 | 246.6 | 0.0213 | 17.20 | 6,069 | 6.54 | 0.0948 | 0.0292 |
| | 246.6 | .0213 | 17.20 | 8,102 | 7.34 | .1265 | .0288 |
| | 246.6 | .0213 | 17.20 | 10,115 | 7.40 | .1580 | .0284 |
| | 246.6 | .0213 | 17.20 | 12,138 | 6.48 | .1895 | .0278 |
| | 246.6 | .0213 | 17.20 | 14,171 | 5.09 | .2213 | .0275 |
| | 246.4 | .0213 | 17.18 | 16,194 | 3.36 | .2529 | .0273 |
| | 246.6 | .0213 | 17.20 | 18,207 | 1.20 | .2843 | .0266 |
| 15 | 247.9 | 0.0212 | 19.37 | 6,079 | 6.51 | 0.0897 | 0.0209 |
| | 247.9 | .0212 | 19.37 | 8,082 | 7.48 | .1193 | .0206 |
| | 247.9 | .0212 | 19.37 | 10,115 | 7.83 | .1493 | .0205 |
| | 248.1 | .0212 | 19.38 | 12,138 | 7.76 | .1792 | .0204 |
| | 248.1 | .0212 | 19.38 | 14,161 | 7.00 | .2091 | .0203 |
| | 248.1 | .0212 | 19.38 | 16,184 | 5.49 | .2389 | .0201 |
| | 248.1 | .0212 | 19.38 | 18,197 | 3.60 | .2686 | .0202 |
| 20 | 248.1 | 0.0212 | 20.73 | 6,089 | 6.32 | 0.0870 | 0.0164 |
| | 248.1 | .0212 | 20.73 | 8,112 | 7.51 | .1158 | .0158 |
| | 248.1 | .0212 | 20.73 | 10,115 | 8.37 | .1444 | .0153 |
| | 248.1 | .0212 | 20.73 | 12,128 | 8.51 | .1732 | .0153 |
| | 248.3 | .0211 | 20.75 | 14,161 | 7.89 | .2022 | .0153 |
| | 248.1 | .0212 | 20.73 | 16,184 | 6.93 | .2311 | .0154 |
| | 248.3 | .0211 | 20.75 | 18,207 | 5.40 | .2600 | .0154 |



TABLE VI - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

NOZZLE G

[Inlet-gas temperature, 1000° F; inlet-gas pressure,
95 lb/sq in. gage]

820

| Pressure ratio | Air mass flow (lb/hr) | Fuel-air ratio | Horsepower available from isentropic expansion | Turbine speed (rpm) | Brake horsepower | Blade-jet speed ratio | Gas density in turbine case (lb/cu ft) |
|----------------|-----------------------|----------------|--|---------------------|------------------|-----------------------|--|
| 8 | 977.6 | 0.0138 | 62.56 | 6,039 | 20.50 | 0.0979 | 0.0334 |
| | 976.2 | .0138 | 62.47 | 8,122 | 23.82 | .1316 | .0334 |
| | 977.6 | .0138 | 62.56 | 10,115 | 25.60 | .1639 | .0332 |
| | 977.6 | .0138 | 62.56 | 12,138 | 26.08 | .1967 | .0340 |
| | 977.6 | .0138 | 62.56 | 14,191 | 25.30 | .2300 | .0341 |
| | 977.6 | .0138 | 62.56 | 16,204 | 23.50 | .2626 | .0342 |
| | 977.6 | .0138 | 62.56 | 18,227 | 20.42 | .2954 | .0340 |
| 10 | 977.6 | 0.0138 | 67.42 | 6,069 | 22.50 | 0.0948 | 0.0275 |
| | 976.9 | .0138 | 67.37 | 8,122 | 26.63 | .1268 | .0268 |
| | 976.9 | .0138 | 67.37 | 10,115 | 28.67 | .1580 | .0266 |
| | 976.9 | .0138 | 67.37 | 12,128 | 29.38 | .1894 | .0273 |
| | 976.7 | .0138 | 67.36 | 14,151 | 28.73 | .2210 | .0274 |
| | 976.0 | .0138 | 67.31 | 16,194 | 26.95 | .2529 | .0276 |
| | 977.6 | .0138 | 67.42 | 18,217 | 24.25 | .2845 | .0275 |
| 15 | 976.5 | 0.0138 | 75.43 | 6,069 | 23.84 | 0.0896 | 0.0191 |
| | 976.5 | .0138 | 75.43 | 8,112 | 28.82 | .1197 | .0189 |
| | 978.1 | .0138 | 75.55 | 10,125 | 31.83 | .1495 | .0189 |
| | 977.2 | .0138 | 75.48 | 12,138 | 33.24 | .1792 | .0191 |
| | 976.5 | .0138 | 75.43 | 14,171 | 32.88 | .2092 | .0193 |
| | 977.2 | .0138 | 75.48 | 16,174 | 31.50 | .2388 | .0196 |
| | 977.2 | .0138 | 75.48 | 18,207 | 29.34 | .2688 | .0199 |
| 20 | 976.5 | 0.0138 | 80.64 | 6,069 | 24.90 | 0.0867 | 0.0153 |
| | 977.2 | .0138 | 80.70 | 8,122 | 30.06 | .1160 | .0156 |
| | 976.5 | .0138 | 80.64 | 10,125 | 33.43 | .1446 | .0154 |
| | 977.2 | .0138 | 80.70 | 12,158 | 35.26 | .1736 | .0152 |
| | 976.5 | .0138 | 80.64 | 14,161 | 35.61 | .2022 | .0152 |
| | 978.1 | .0138 | 80.76 | 16,204 | 34.02 | .2314 | .0158 |
| | 977.2 | .0138 | 80.70 | 18,177 | 32.23 | .2596 | .0162 |



TABLE VII - SUMMARY OF EFFICIENCY DATA FOR MARK 25 TURBINE

WITH NOZZLE H

[Inlet-gas temperature, 1000° F; inlet-gas pressure,
95 lb/sq in. gage]

| Pressure ratio | Air mass flow (lb/hr) | Fuel-air ratio | Horsepower available from isentropic expansion | Turbine speed (rpm) | Brake horsepower | Blade-jet speed ratio | Gas density in turbine case (lb/cu ft) |
|----------------|-----------------------|----------------|--|---------------------|------------------|-----------------------|--|
| 8 | 1143.7 | 0.0143 | 73.23 | 6,069 | 28.24 | 0.0984 | 0.0322 |
| | 1143.7 | .0143 | 73.23 | 8,072 | 33.25 | .1308 | .0331 |
| | 1143.7 | .0143 | 73.23 | 10,054 | 36.78 | .1630 | .0332 |
| | 1143.7 | .0143 | 73.23 | 12,189 | 38.56 | .1976 | .0338 |
| | 1143.7 | .0143 | 73.23 | 14,161 | 38.27 | .2295 | .0345 |
| | 1143.7 | .0143 | 73.23 | 16,164 | 36.54 | .2620 | .0346 |
| | 1143.7 | .0143 | 73.23 | 18,187 | 33.26 | .2948 | .0347 |
| 10 | 1152.0 | 0.0141 | 79.48 | 6,069 | 29.62 | 0.0948 | 0.0257 |
| | 1152.0 | .0141 | 79.48 | 8,112 | 35.50 | .1267 | .0265 |
| | 1152.0 | .0141 | 79.48 | 10,105 | 39.49 | .1578 | .0273 |
| | 1152.0 | .0141 | 79.48 | 12,138 | 41.00 | .1895 | .0278 |
| | 1144.1 | .0142 | 78.94 | 14,171 | 41.10 | .2213 | .0284 |
| | 1144.1 | .0142 | 78.94 | 16,164 | 39.63 | .2524 | .0288 |
| | 1152.0 | .0141 | 79.48 | 18,187 | 36.80 | .2840 | .0293 |
| 15 | 1144.1 | 0.0142 | 88.43 | 6,089 | 31.02 | 0.0899 | 0.0198 |
| | 1144.1 | .0142 | 88.43 | 8,122 | 37.87 | .1199 | .0197 |
| | 1144.1 | .0142 | 88.43 | 10,135 | 42.35 | .1496 | .0201 |
| | 1144.1 | .0142 | 88.43 | 12,138 | 45.16 | .1792 | .0200 |
| | 1144.1 | .0142 | 88.43 | 14,171 | 46.84 | .2092 | .0204 |
| | 1144.1 | .0142 | 88.43 | 16,184 | 46.13 | .2389 | .0214 |
| | 1144.1 | .0142 | 88.43 | 18,217 | 43.28 | .2689 | .0224 |
| 19 | 1144.5 | 0.0142 | 93.51 | 6,099 | 32.00 | 0.0876 | 0.0177 |
| | 1144.5 | .0142 | 93.51 | 8,092 | 38.72 | .1162 | .0180 |
| | 1144.5 | .0142 | 93.51 | 10,125 | 43.81 | .1454 | .0182 |
| | 1144.5 | .0142 | 93.51 | 12,148 | 47.00 | .1745 | .0184 |
| | 1144.5 | .0142 | 93.51 | 14,171 | 48.75 | .2035 | .0184 |
| | 1144.5 | .0142 | 93.51 | 16,184 | 48.37 | .2324 | .0192 |
| | 1144.5 | .0142 | 93.51 | 18,217 | 46.35 | .2616 | .0197 |



TABLE VIII - SUMMARY OF CLEARANCE INDICATOR DATA FOR THERMAL
EXPANSION OF MARK 25 TURBINE

[Inlet-gas pressure, 95 lb/sq in. gage]

| Pressure ratio | Turbine speed (rpm) | Inlet- gas tem- perature (°F) | Outlet-gas temperature (°F) | Change in clearance (in.) | | | |
|-------------------|---------------------------|--|-----------------------------------|------------------------------|---------------|--------------|---------------|
| | | | | Axial | | Radial | |
| | | | | Left gage | Right gage | Left gage | Right gage |
| ----- 10 | 0 | 73 | 88 | 0 | 0 | 0 | 0 |
| | 6,069 | 502 | 281 | -.0120 | -.0130 | -.0040 | -.0035 |
| | 6,069 | 748 | 465 | -.0130 | -.0145 | -.0050 | -.0050 |
| | 6,069 | 1003 | 662 | -.0140 | -.0160 | -.0070 | -.0065 |
| | 6,069 | 1250 | 851 | -.0140 | -.0165 | -.0075 | -.0020 |
| | 6,069 | 1485 | 1048 | -.0090 | -.0090 | -.0090 | 0 |
| ----- 15 | 0 | 70 | 80 | 0 | 0 | 0 | 0 |
| | 6,069 | 495 | 261 | -.0090 | -.0070 | -.0030 | -.0040 |
| | 6,069 | 739 | 443 | -.0150 | -.0115 | (a) | -.0040 |
| | 6,069 | 993 | 620 | -.0140 | -.0130 | (a) | -.0035 |
| | 6,069 | 1248 | 810 | -.0140 | -.0100 | (a) | 0 |
| | 6,069 | 1507 | 1004 | -.0110 | -.0075 | (a) | .0040 |
| ----- 20 | 0 | 68 | 75 | 0 | 0 | 0 | 0 |
| | 6,069 | 503 | 253 | -.0080 | -.0085 | -.0050 | -.0030 |
| | 6,069 | 753 | 438 | -.0105 | -.0100 | -.0080 | -.0035 |
| | 6,069 | 1010 | 634 | -.0120 | -.0110 | -.0100 | -.0045 |
| | 6,069 | 1251 | 817 | -.0135 | -.0095 | -.0110 | -.0020 |
| | 6,069 | 1498 | 987 | -.0120 | -.0090 | -.0130 | .0030 |
| ----- 15 | 0 | 70 | 82 | 0 | 0 | 0 | 0 |
| | 12,138 | 489 | 190 | -.0060 | -.0080 | -.0050 | -.0035 |
| | 12,138 | 753 | 374 | -.0090 | -.0110 | -.0075 | -.0045 |
| | 12,138 | 998 | 542 | -.0130 | -.0135 | -.0095 | -.0055 |
| | 12,138 | 1256 | 712 | -.0150 | -.0145 | -.0110 | -.0020 |
| | 12,138 | 1505 | 883 | -.0130 | -.0145 | -.0090 | .0010 |
| ----- 20 | 0 | 90 | 90 | 0 | 0 | 0 | 0 |
| | 12,138 | 497 | 205 | -.0050 | -.0070 | -.0030 | -.0025 |
| | 12,138 | 750 | 370 | -.0080 | -.0085 | -.0055 | -.0040 |
| | 12,138 | 1012 | 524 | -.0100 | -.0110 | -.0080 | -.0050 |
| | 12,138 | 1255 | 690 | -.0125 | -.0125 | -.0090 | (a) |
| | 12,138 | 1508 | 864 | -.0150 | -.0115 | -.0070 | (a) |

^a Indicator failed.

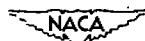


TABLE IX - SUMMARY OF CLEARANCE-INDICATOR DATA FOR CENTRIFUGAL

EXPANSION OF MARK 25 TURBINE

[Room temperature and pressure]

| Turbine speed (rpm) | Radial expansion | |
|---------------------------|------------------|---------------|
| | Left gage | Right gage |
| 0 | 0 | 0 |
| 6,069 | .0005 | .0005 |
| 8,092 | .0010 | .0010 |
| 10,115 | .0010 | .0010 |
| 12,138 | .0015 | .0015 |
| 14,161 | .0025 | .0025 |
| 16,184 | .0040 | .0040 |
| 18,207 | .0045 | .0045 |



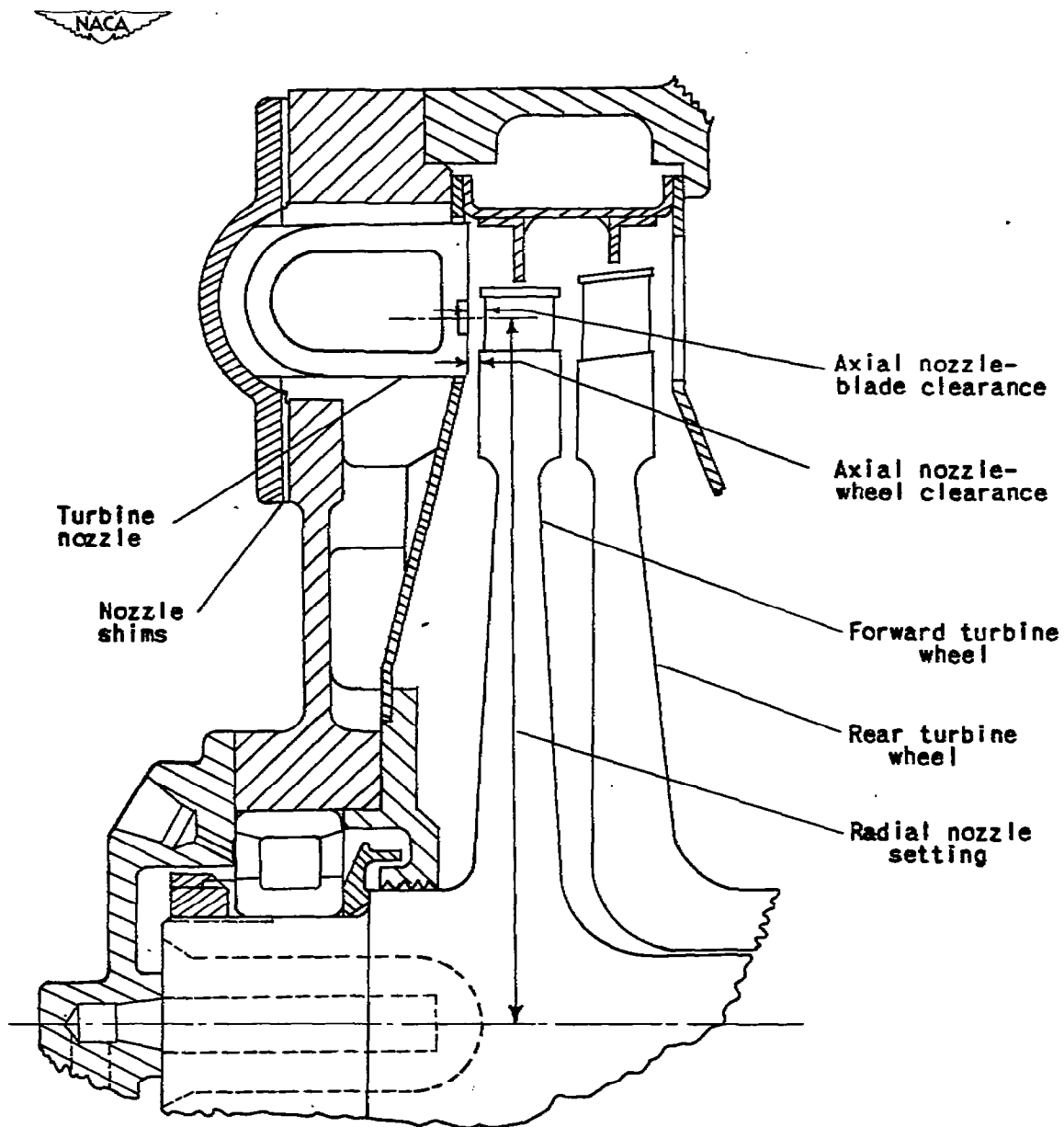
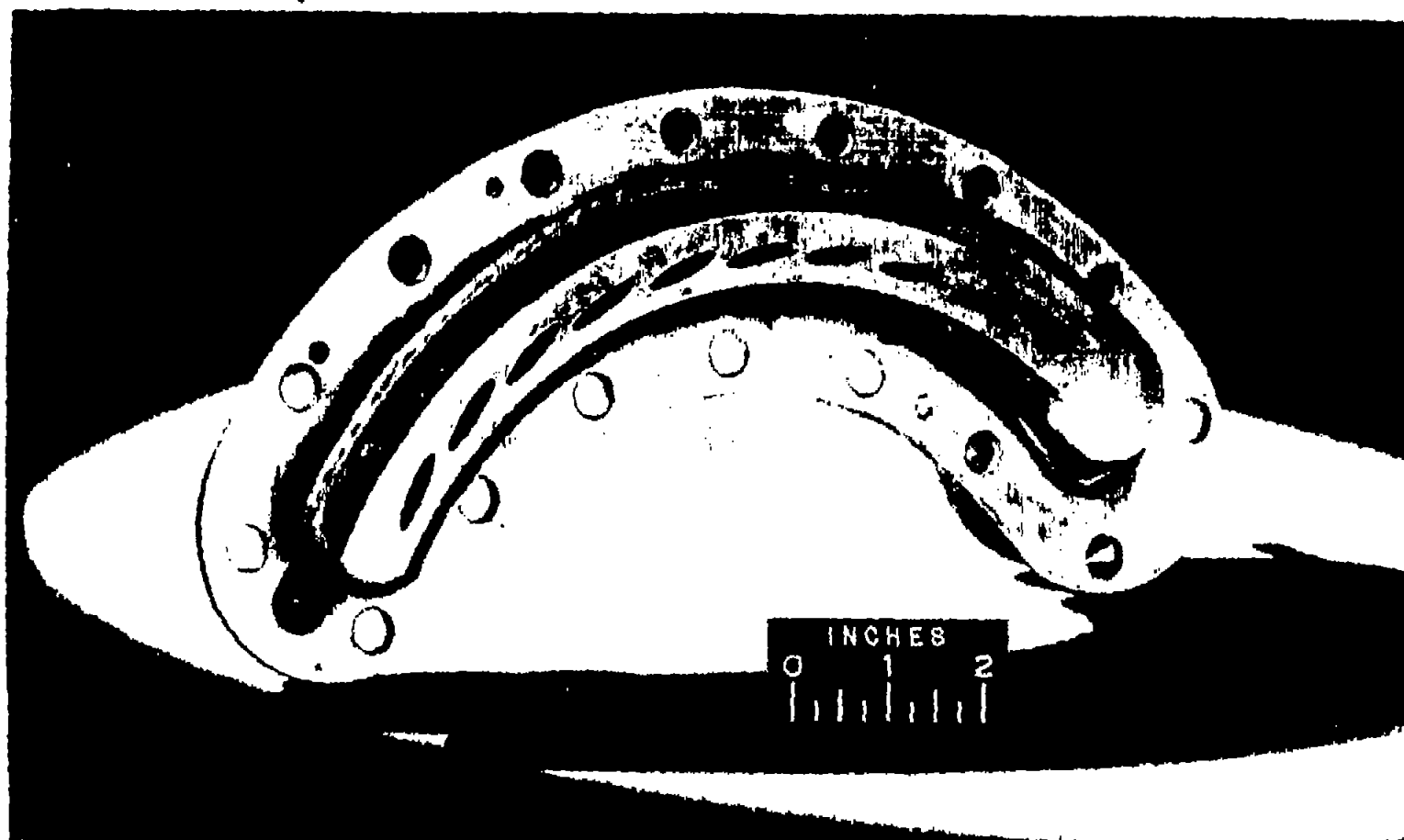


Figure 1. - Sketch of nozzle-and-turbine-wheel assembly for Mark 25 torpedo power plant.



(a) Reamed nozzle E.



C-18948
10-2-48

Figure 2. - Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.

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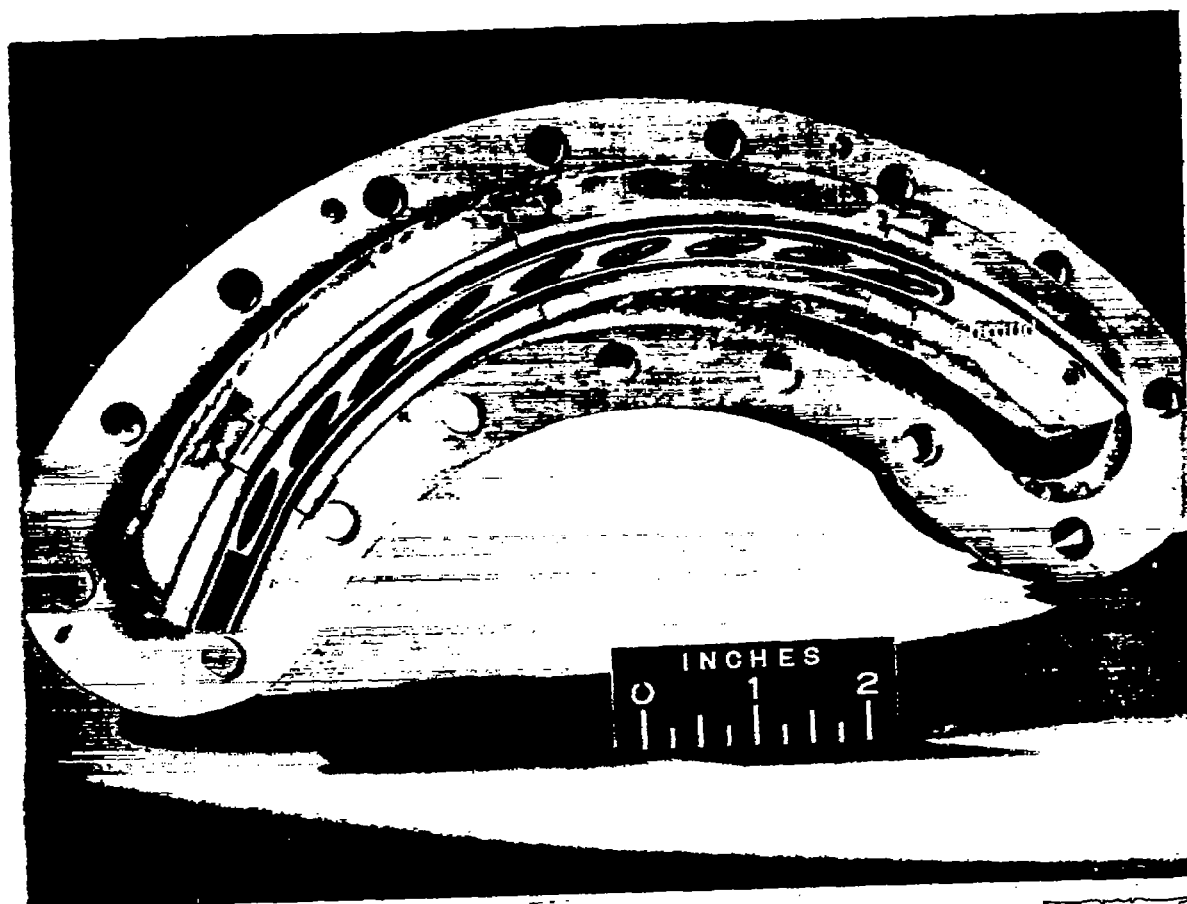
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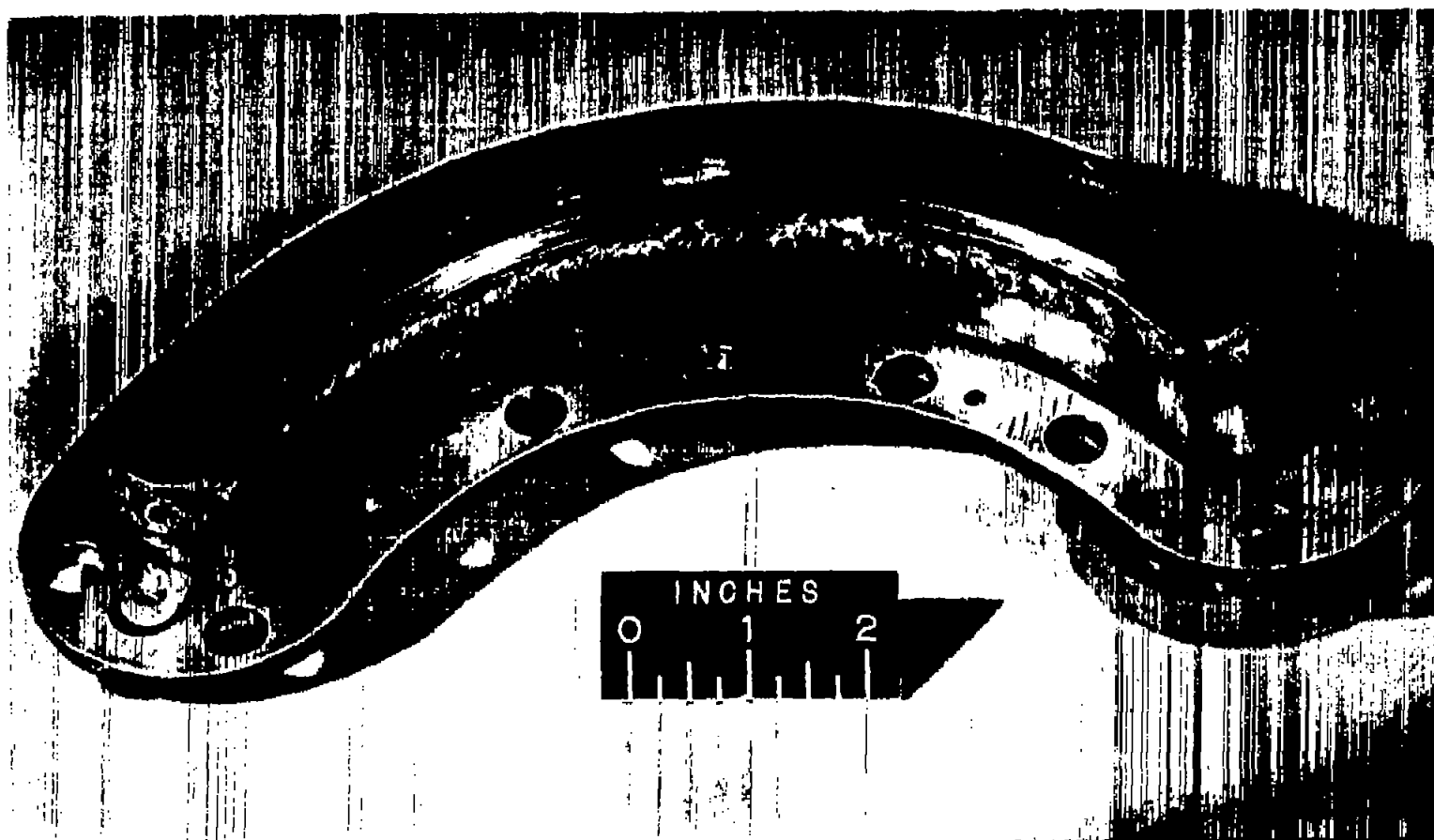
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(b) Reamed, shrouded nozzle G.

NACA
C-15948
10-2-46

Figure 2. - Continued. Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.



(c) Cast nozzle H.

NACA
C-18384
4-23-47

Figure 2. - Concluded. Outlet face of various nozzles designed for turbine of Mark 25 torpedo power plant.

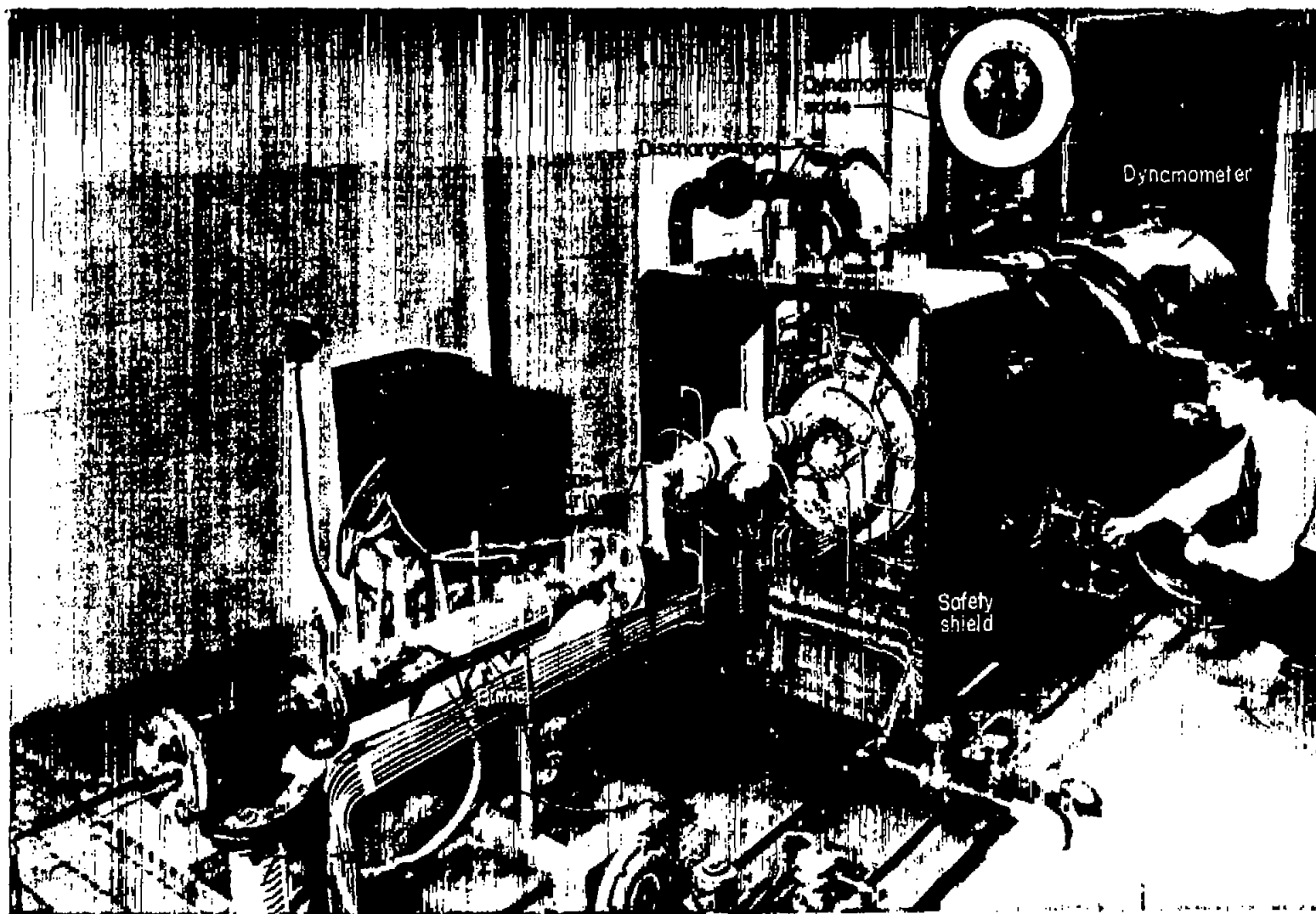


Figure 3. - Setup for investigation of turbine of Mark 25 torpedo power plant.

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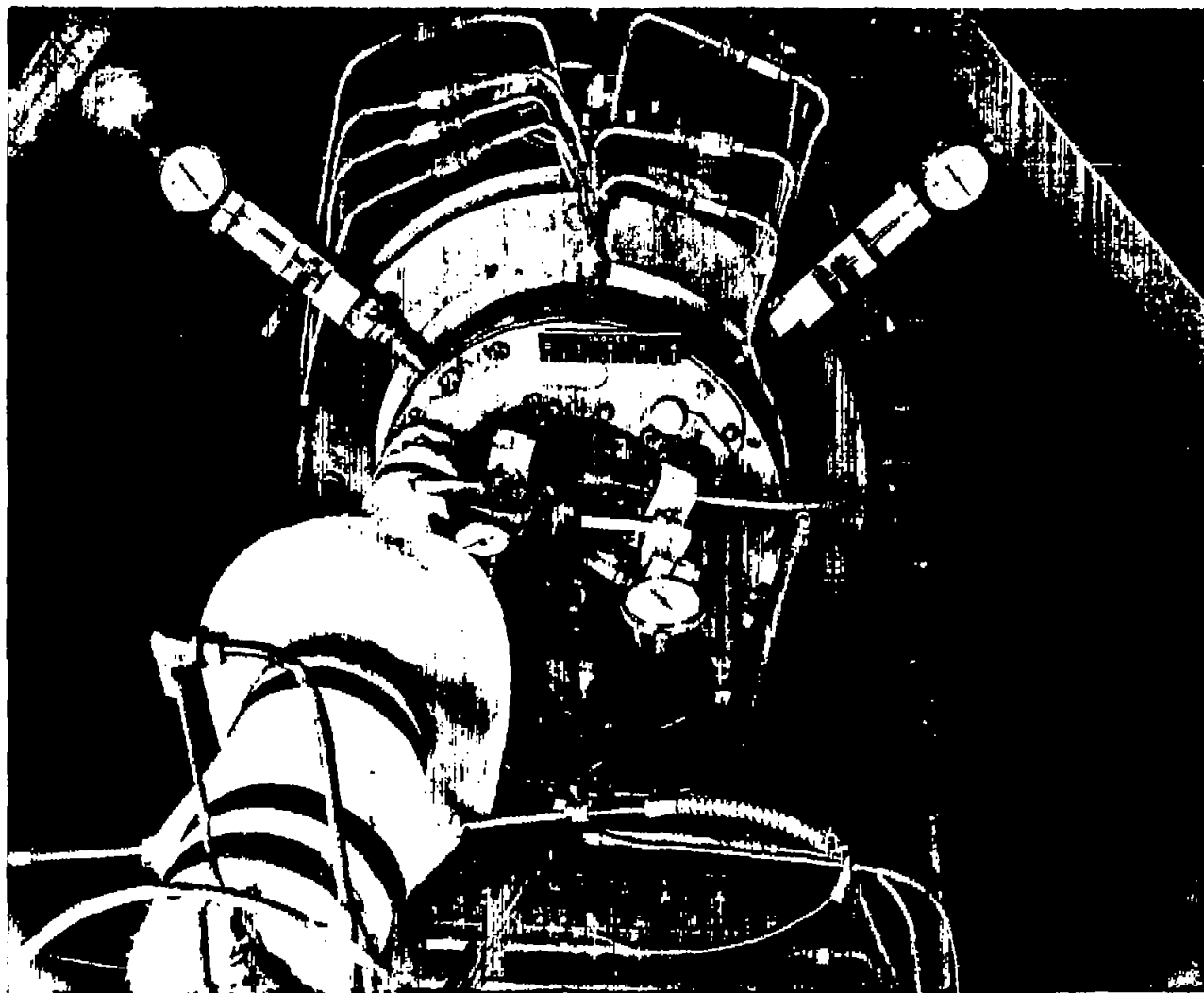
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C-17837
2-7-47

Figure 4. - Clearance indicators installed on turbine unit to measure changes in radial nozzle setting and axial nozzle-wheel clearance.

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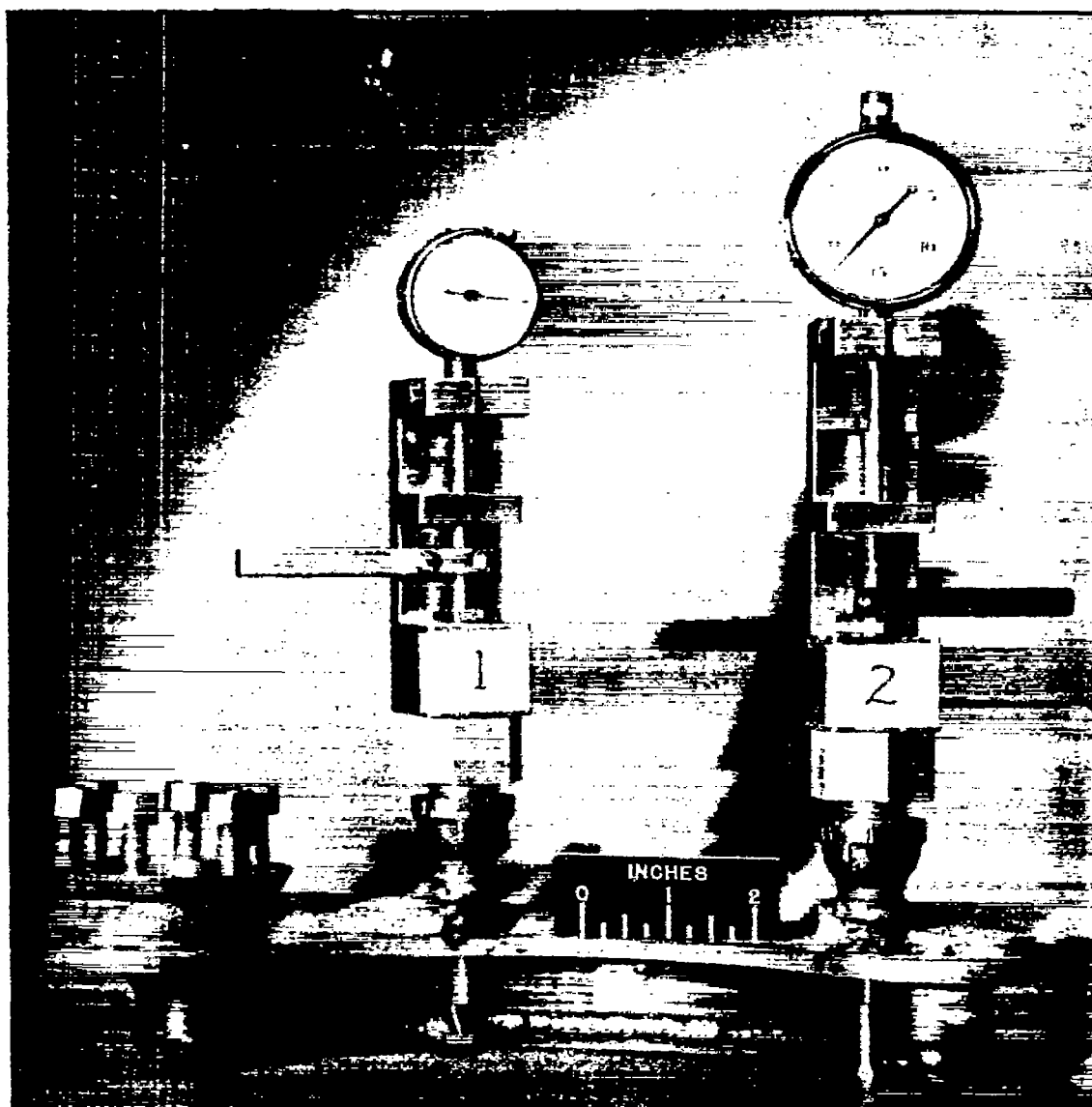
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NACA
C-18047
3-4-47

Figure 5. - Axial-clearance indicators installed on nozzle.

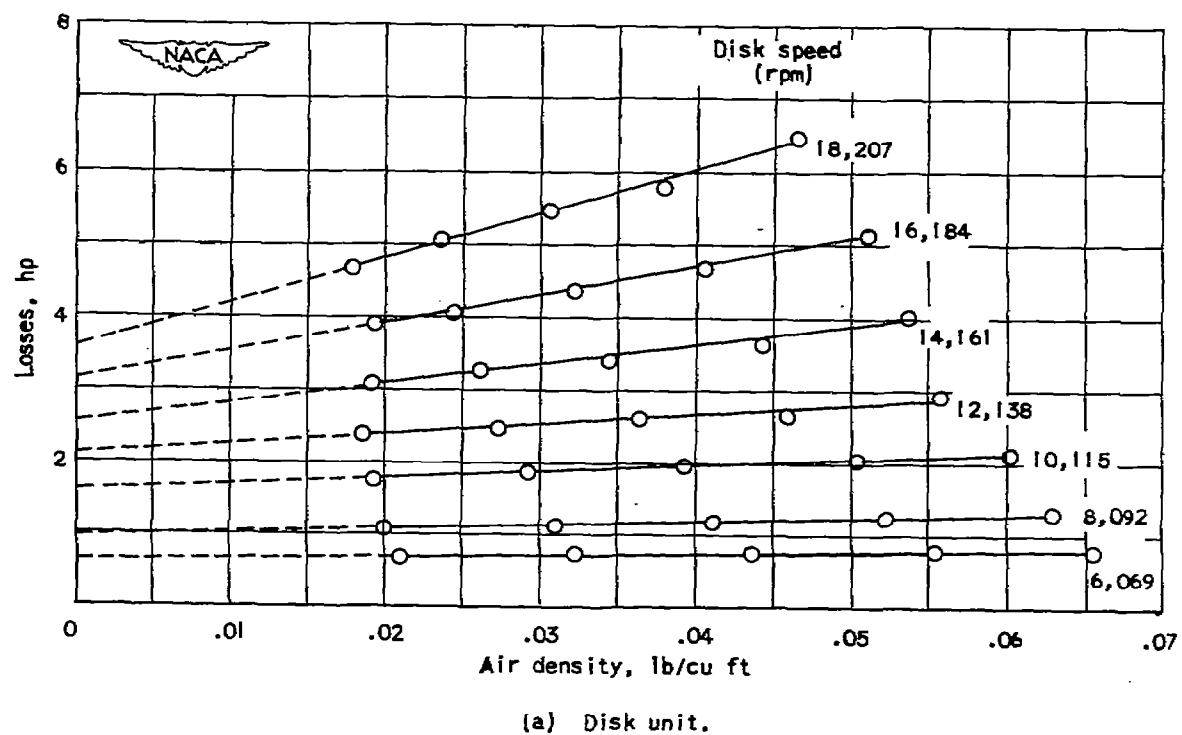
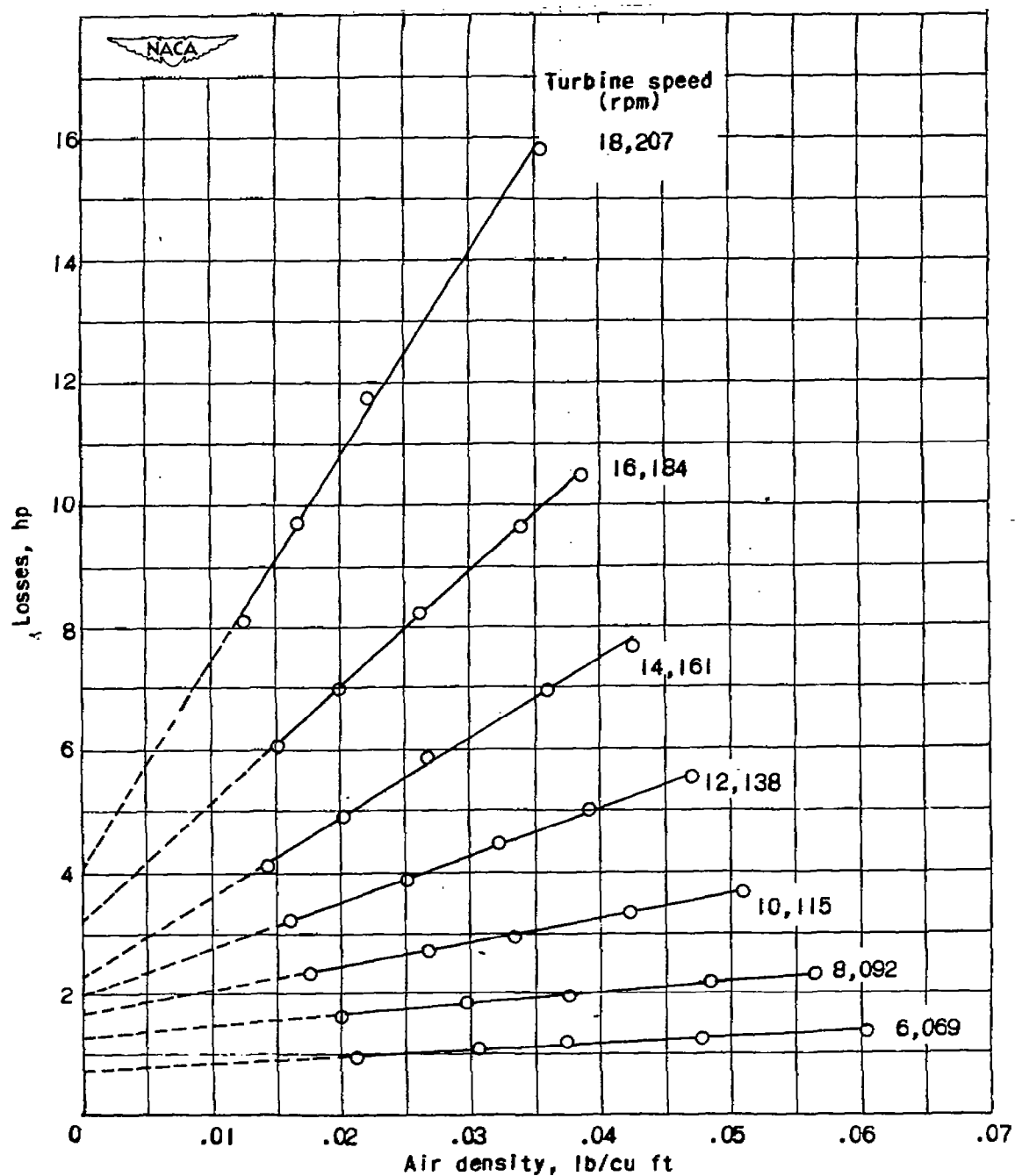
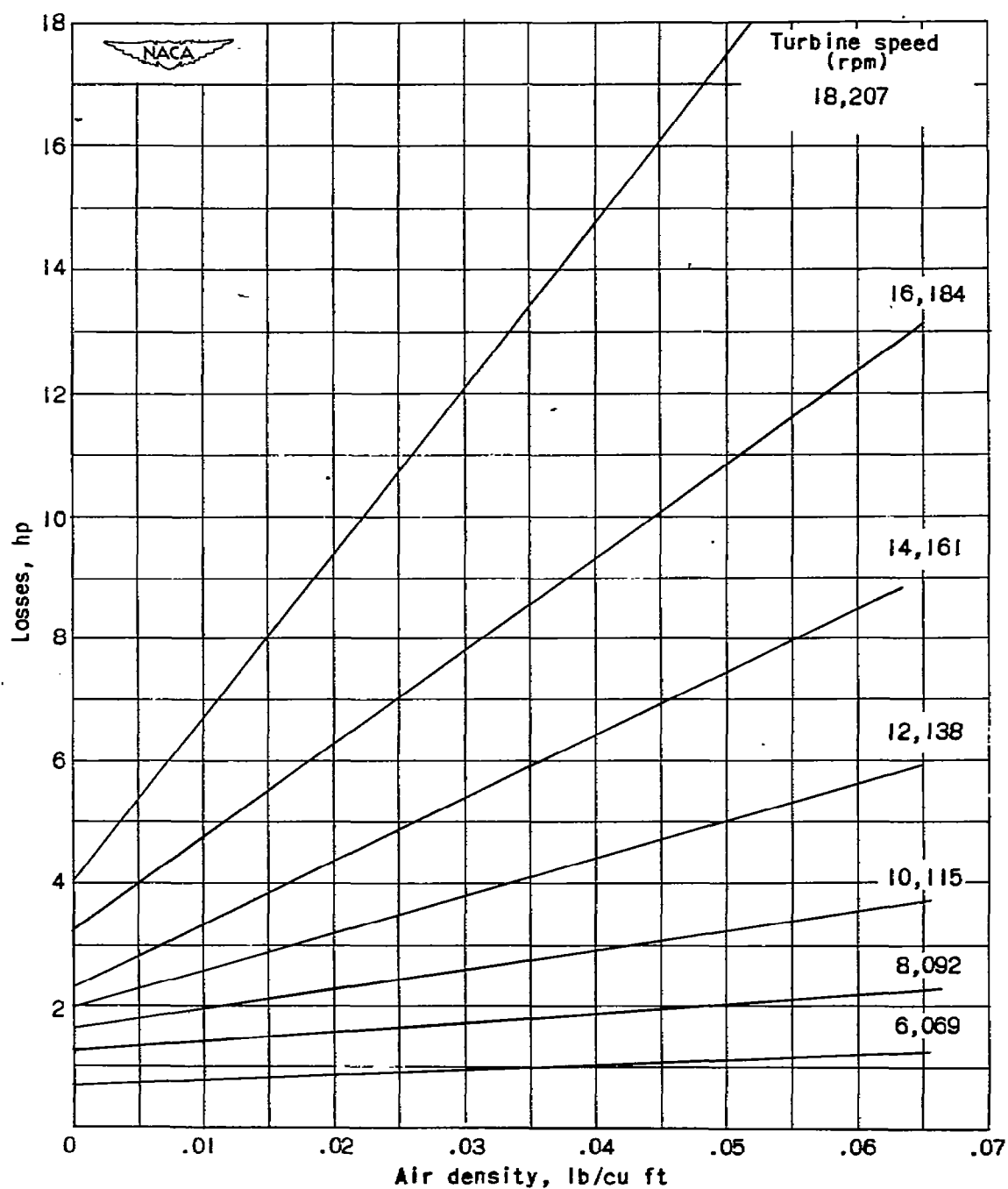


Figure 6. - Variation of windage and mechanical losses with air density and speed.



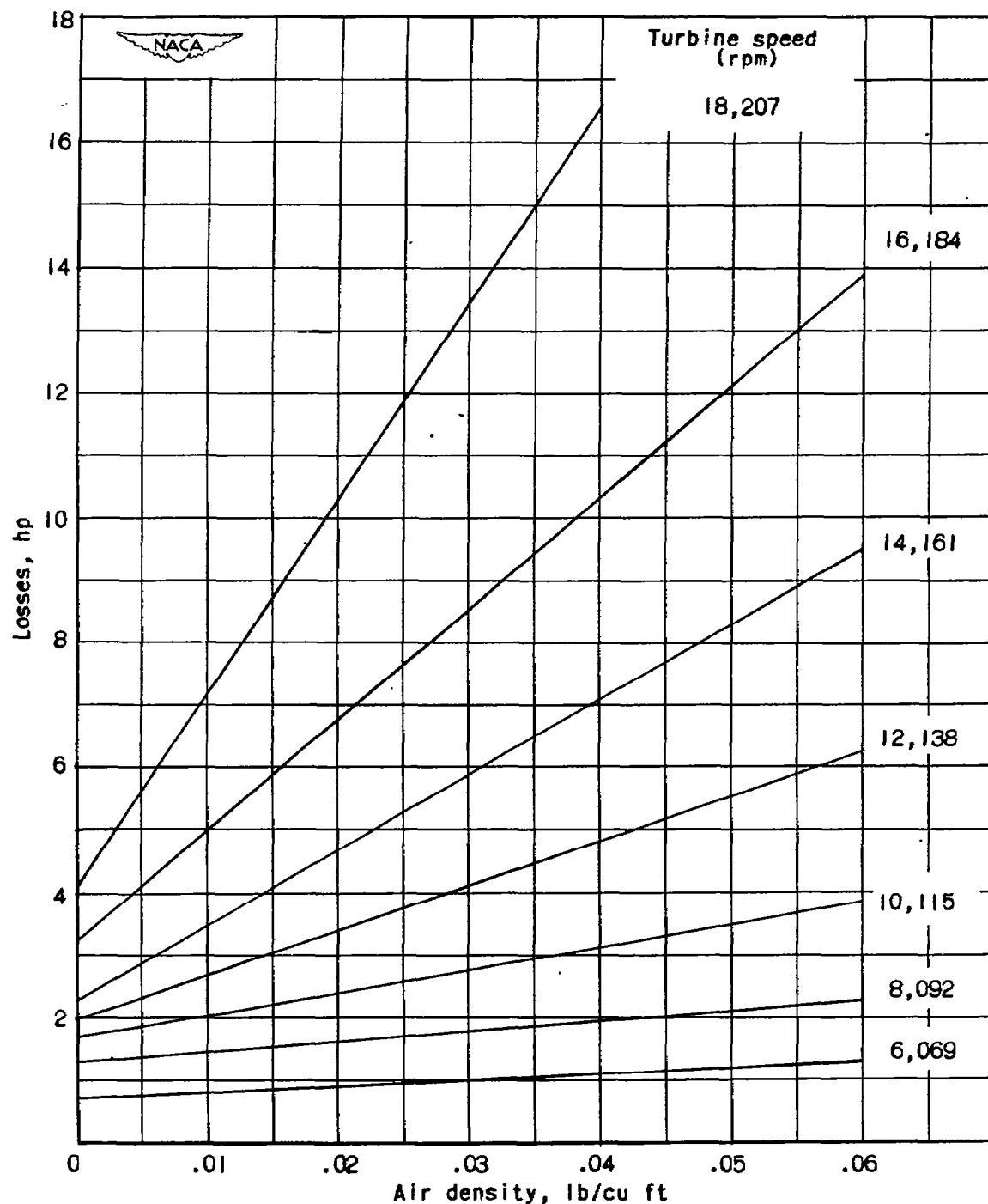
(b) Turbine wheels with blades.

Figure 6. - Concluded. Variation of windage and mechanical losses with air density and speed.



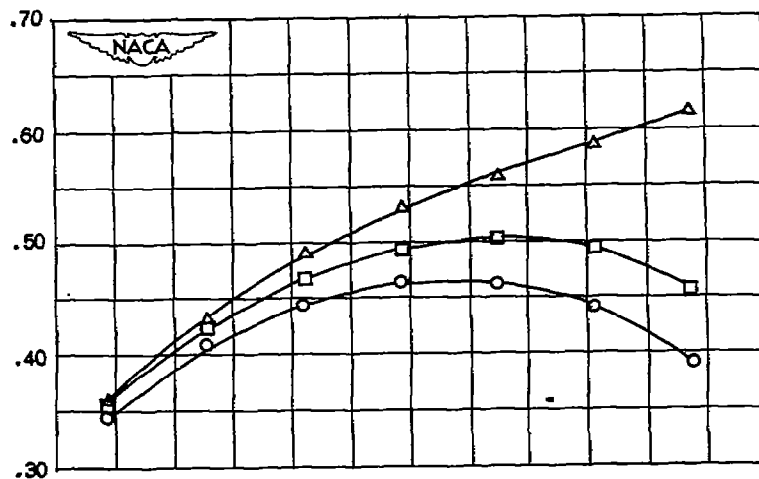
(a) Disk loss plus three-fourths blade loss for nine-port nozzle.

Figure 7. - Variation of calculated windage and mechanical losses with air density and turbine speed for partial gas admission. (Data from fig. 6.)

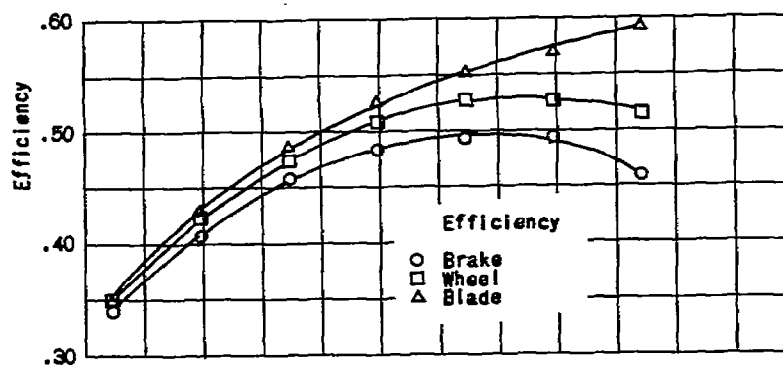


(b) Disk loss plus eleven-twelfths blade loss for three-port nozzle.

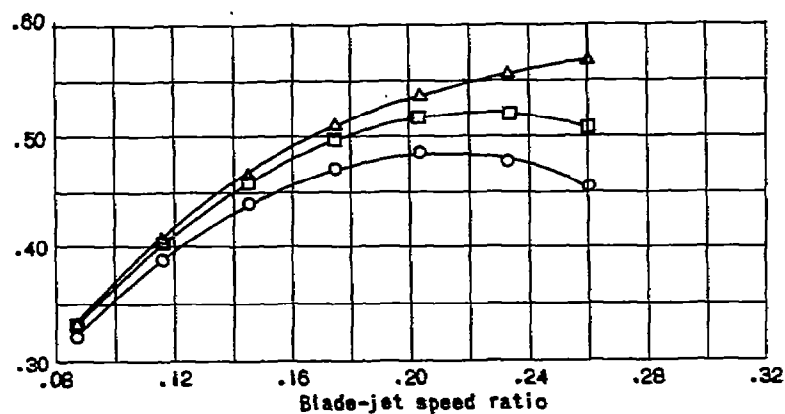
Figure 7. - Concluded. Variation of calculated windage and mechanical losses with air density and turbine speed for partial gas admission. (Data from fig. 6.)



(a) Pressure ratio, 8.



(b) Pressure ratio, 15.



(c) Pressure ratio, 20.

Figure 8. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle A at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

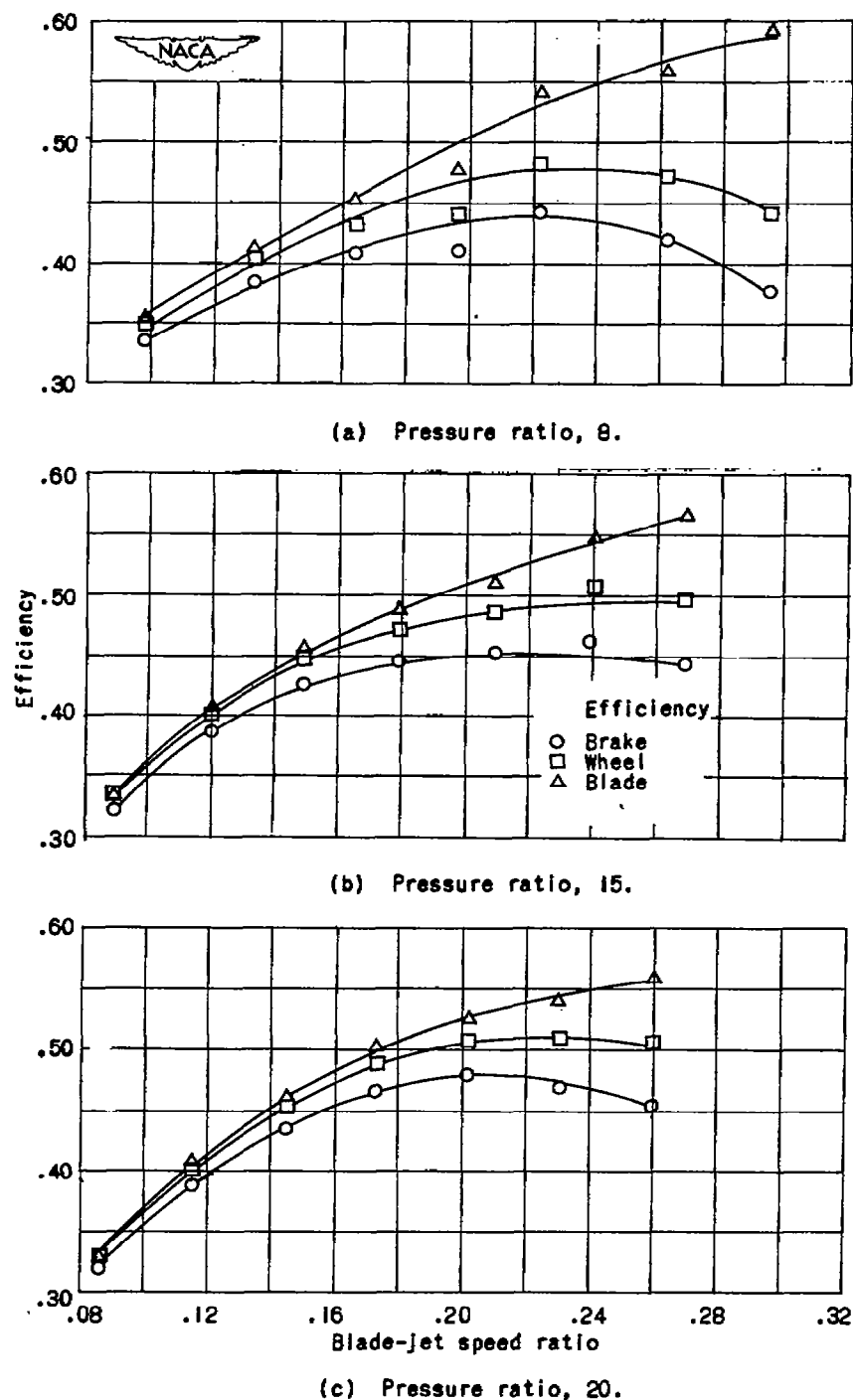


Figure 9. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle E at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

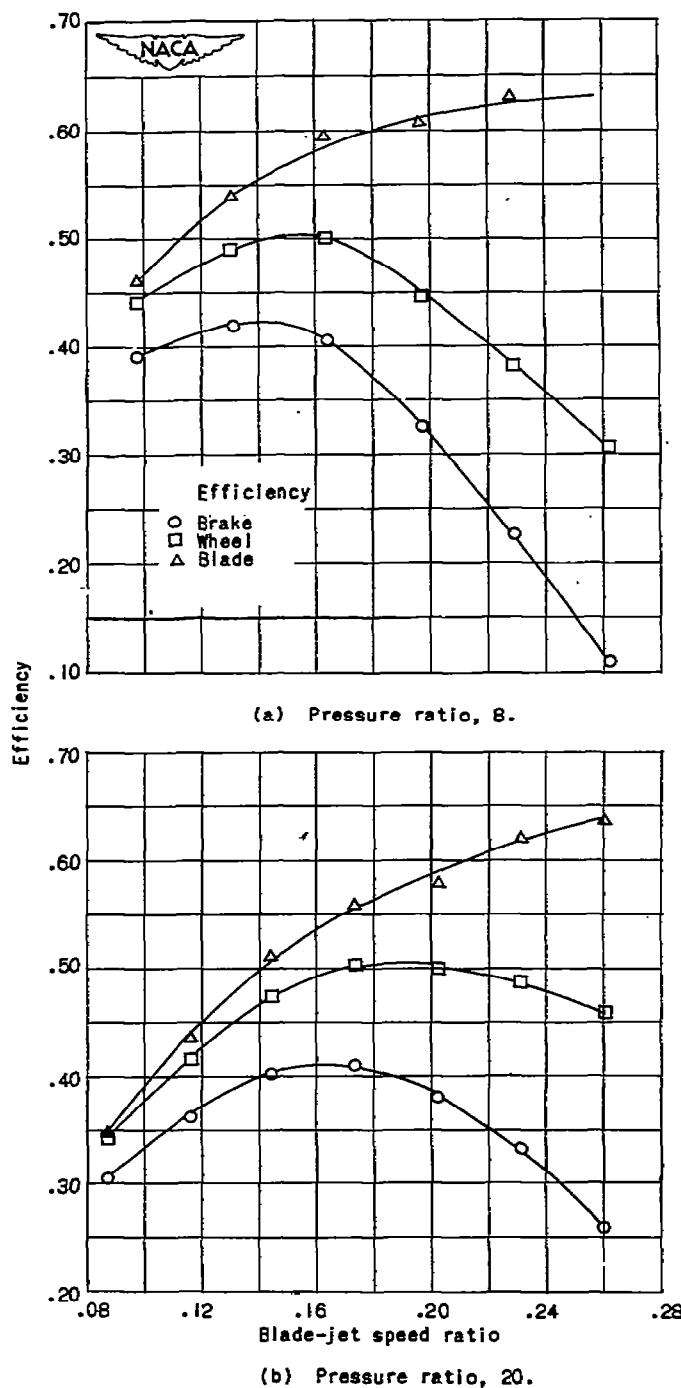


Figure 10. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle F at two pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

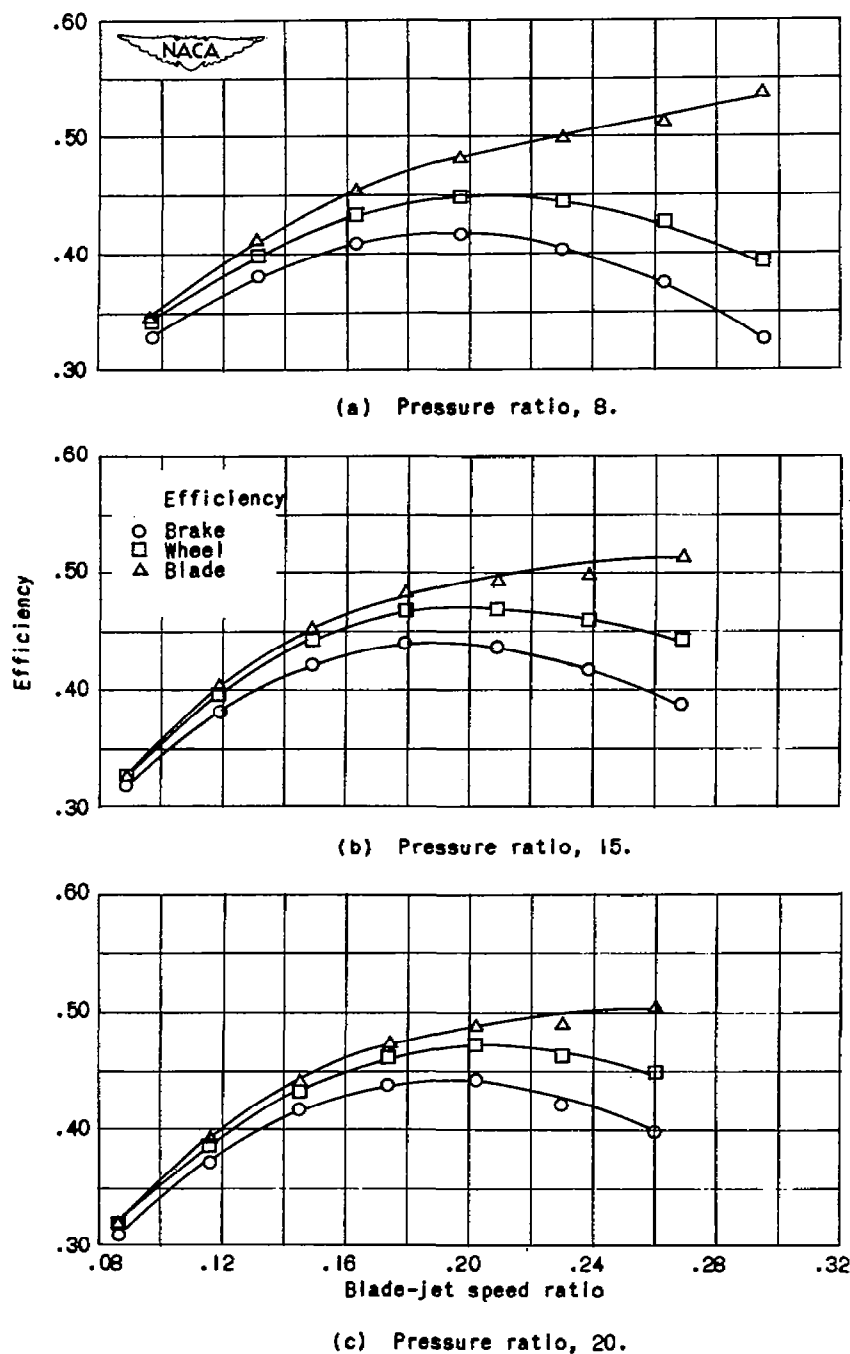


Figure 11. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle G at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

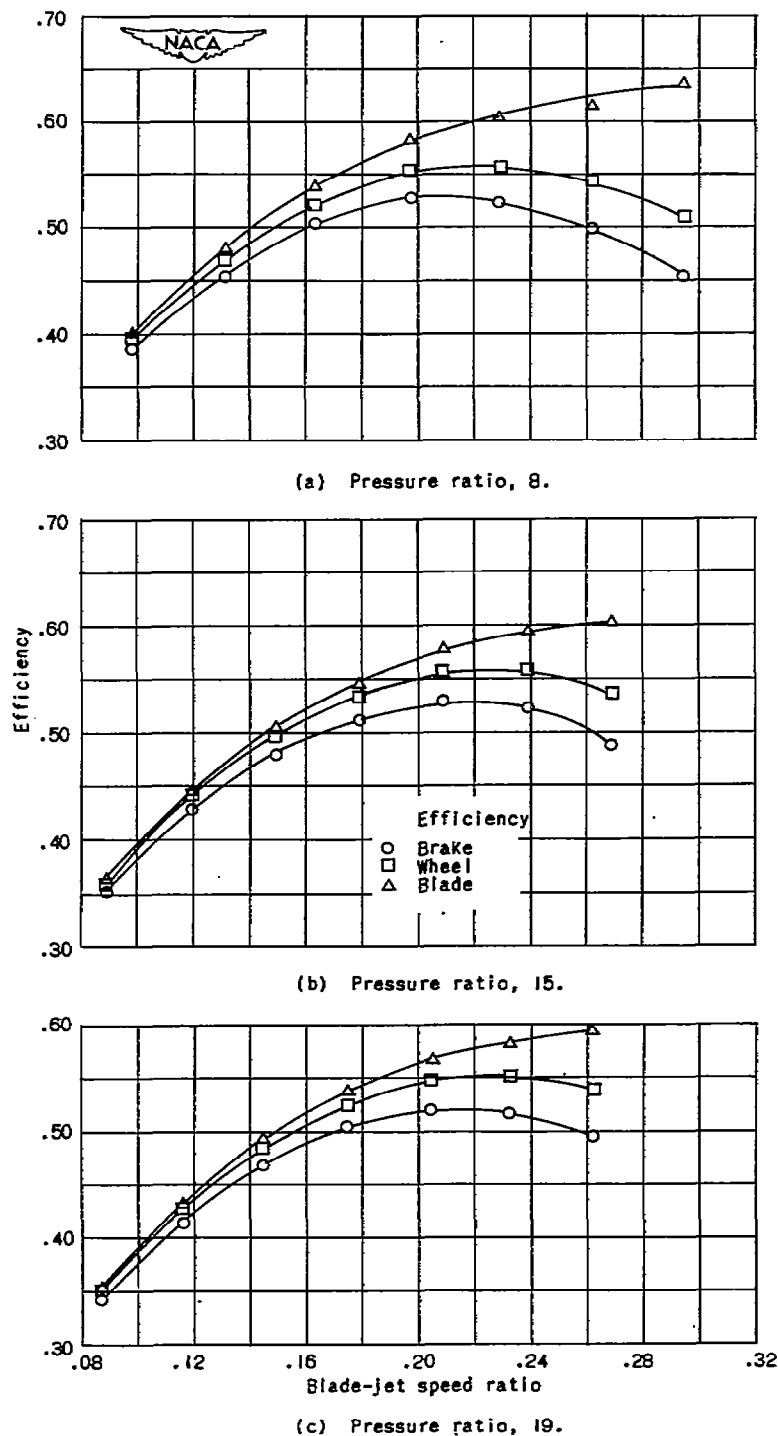
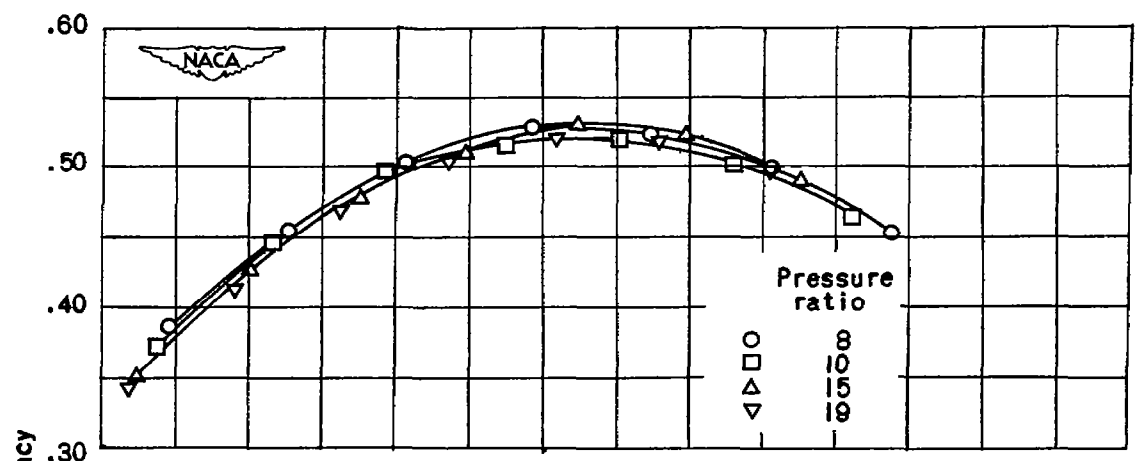
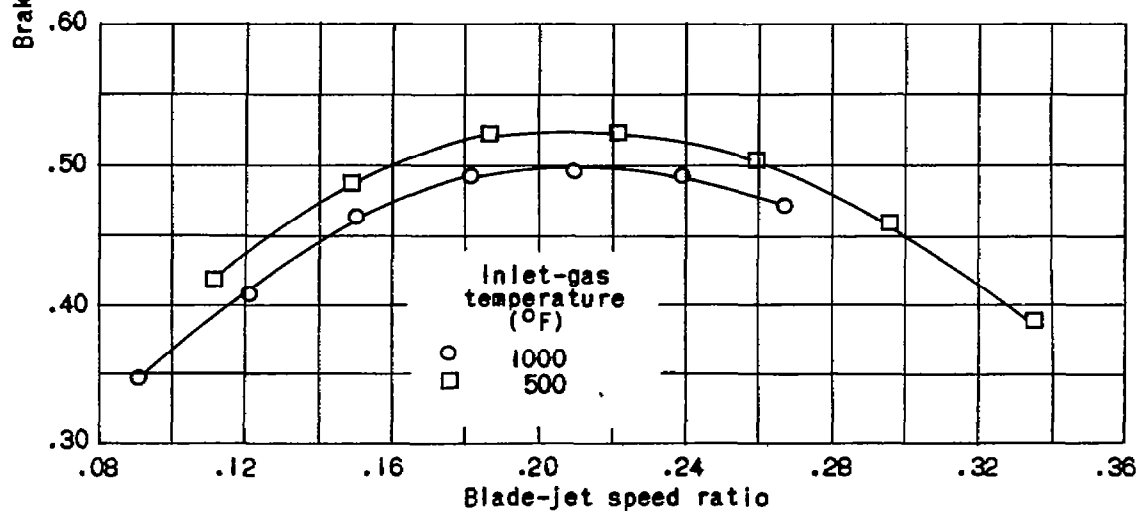


Figure 12. - Variation of power-plant component efficiencies with blade-jet speed ratio for nozzle H at three pressure ratios. Inlet-gas pressure, 95 pounds per square inch gage; inlet-gas temperature, 1000° F; axial nozzle-wheel running clearance, 0.030 inch.

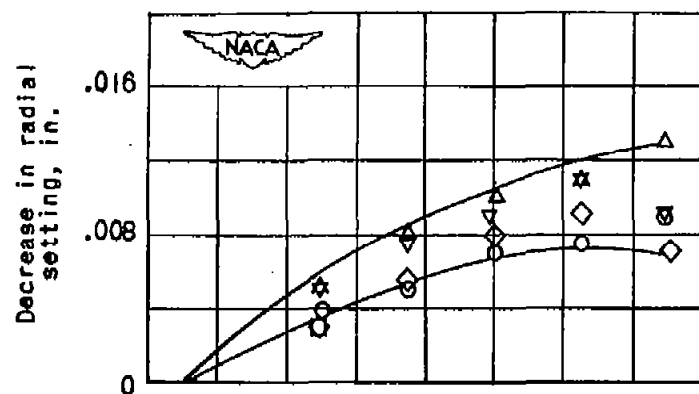


(a) Effect of pressure ratio with nozzle H at 0.030-inch axial nozzle-wheel clearance and inlet-gas temperature of 1000° F.

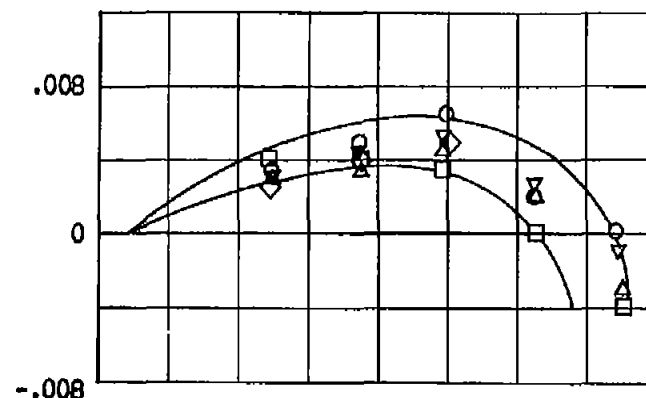


(b) Effect of inlet-gas temperature with nozzle A at 0.060-inch axial nozzle-wheel clearance and pressure ratio of 15.

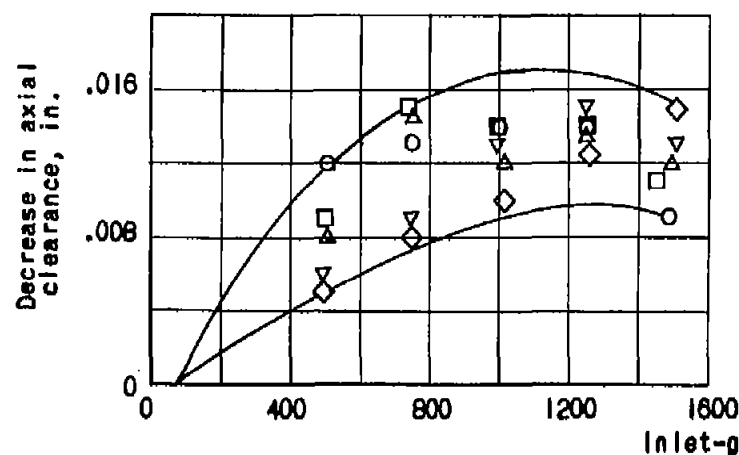
Figure 13. - Variation of turbine brake efficiency with pressure ratio and inlet-gas temperature at inlet-gas pressure of 95 pounds per square inch gage.



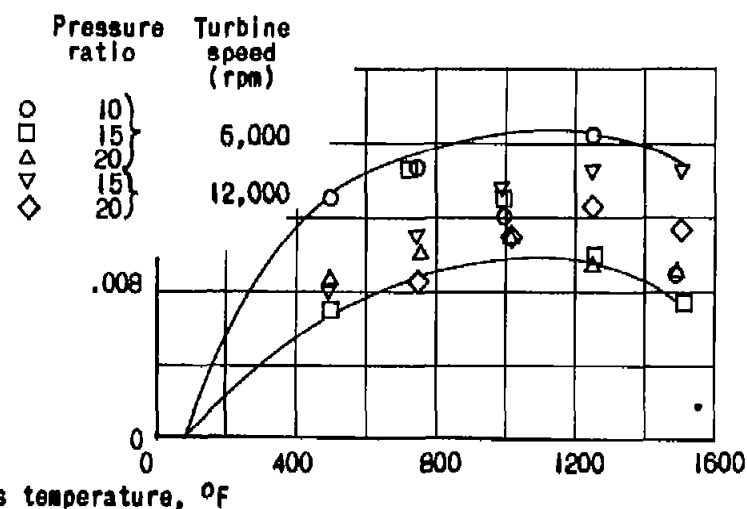
(a) Left radial gage.



(b) Right radial gage.

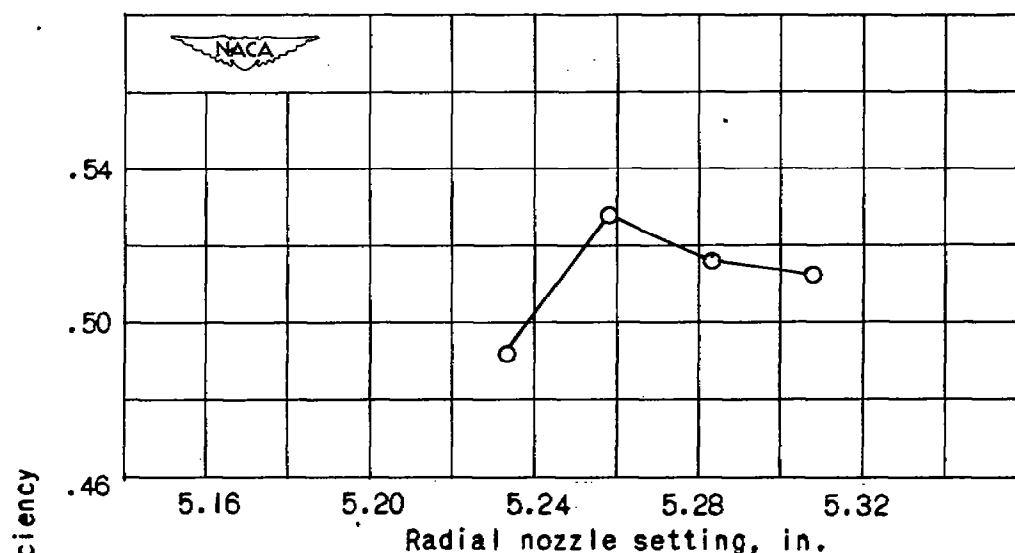


(c) Left axial gage.

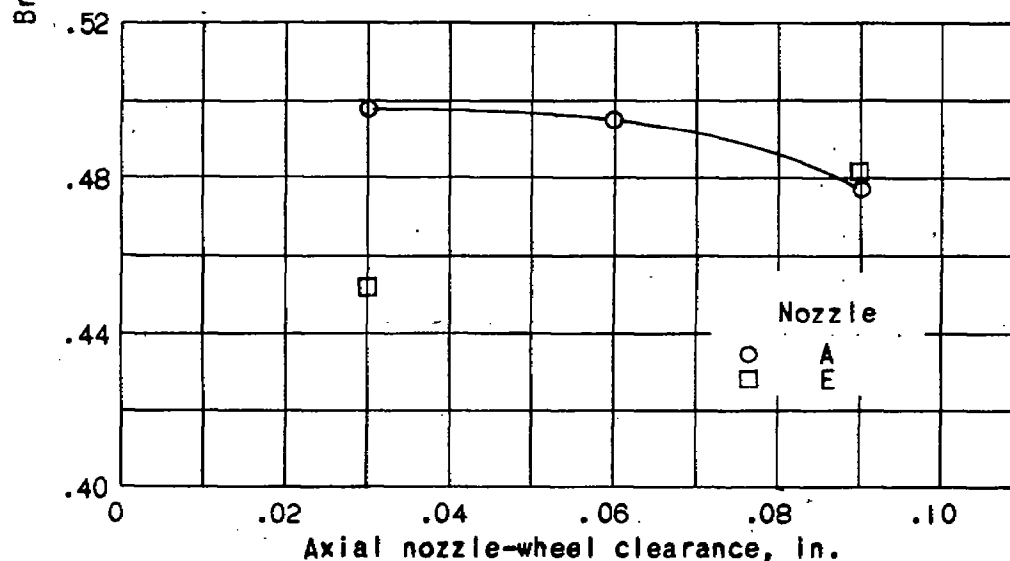


(d) Right axial gage.

Figure 14. - Changes in radial setting and axial clearance from cold clearances with inlet-gas temperature at inlet-gas pressure of 95 pounds per square inch gage.



(a) Effect of radial setting with nozzle H and axial nozzle-wheel clearance of 0.030 inch.



(b) Effect of axial running clearance with radial nozzle setting of 5.283 inches.

Figure 15. - Variation of turbine brake efficiency with radial nozzle setting and axial clearance. Inlet-gas pressure of 95 pounds per square inch gage; inlet-gas temperature of 1000° F; blade-jet speed ratio, 0.21; pressure ratio, 15.

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